AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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Forty Years

WITH this issue, *The Automobile Engineer* completes 40 years of continuous publication. Certain of these years have been difficult ones, more particularly those of the two great wars. Of these the last was of course much the worst, owing to the disorganization, direct and indirect, created by enemy action, shortage of labour and paper supplies, etc. Although not without significance, the attainment of a fortieth birthday of this kind is not the occasion for a very complete survey of past history. That will be better left to the time when we can look back upon the completion of the half century. The present occasion does, however, seem to be one that calls for a brief commemorative note.

An unusual position

For some unaccountable reason, *The Automobile Engineer* continues to occupy, in the precise meaning of the word, a unique position in the world of journalism, in that it is still the only publication of its kind in existence. Neither has there been, at any time during the forty years of its life, any attempt, serious or otherwise, to produce a journal of this type, solely devoted to the design and manufacture of road vehicles. Other than *The Automobile Engineer*, nothing of the sort exists anywhere else in the world. For this reason it serves not only the British industry, but the automobile industries of all the vehicle-producing countries as well. This we know of course by our contacts and circulation. Every other journal, whether trade, technical, scientific, or merely news and general, has one or more rival publications, not only within its country of origin, but in other countries as well. *The Automobile Engineer* is, however, still without a competitor, and this is the more curious in that the industry we serve is the largest in the world. It is the greatest consumer of raw materials such as steel, malleable iron, rubber, glass, and a myriad other commodities. The American automobile industry is still easily ahead in magnitude, but Great Britain is the second largest producer.

In world output figures, the striking feature of motor manufacture is its sustained, and quite phenomenal growth. The number of makers has fallen through the years, but those that remain have increased correspondingly in magnitude and power. During this really stupendous expansion, more particularly in its relation to the small, tentative, and almost experimental beginnings, it is interesting to study the approach of *The Automobile Engineer* at its inception in 1910. We then realised that from the technical or engineering angle, highly important events lay ahead. The fact that no journal then existed to serve these interests, was of course the reason for founding *The Automobile Engineer*.

In the leading article in our issue for June, 1910, No. 1, Volume 1, it is stated that journals devoted to all branches of engineering have not space to deal adequately with automobile design and construction—"we are confident that in producing *The Automobile Engineer* we shall have the support of all those to whom the theory and practice of automobile engineering are of interest and importance." Here there appears to have been some flash of insight into the future, for we continue, "we do not confine ourselves to Great Britain alone but anticipate confidently a welcome in every automobile works throughout the world." How completely this objective has been realized we have commented upon.

Editorial policy

Continuing from our original leader, we state that "our aim is to become the first authority on the building of self-propelled vehicles. We shall give publicity to all research that appears valuable, and shall afford facilities for thrashing out controversial technical subjects. It will also be our endeavour to describe and criticize current designs with a thoroughness hitherto unattempted." We also say with regard to this criticism that "we are able to do this by our knowledge of the subject from an engineer's point of view." While commending the confidence that is implicit in this statement, the same intention could probably have been expressed in rather more becomingly modest terms. However, this is the ambitious course we set ourselves to pursue, and to the best of our ability we have followed the lines then laid down. We say that "we

shall assist engineers towards the production of ever improving automobiles by ever improving means." It is our hope today, that we may perhaps have in fact assisted a little towards these ends. The final line of this somewhat forthright leader, states categorically, "we shall take an important place in the future history of automobilism."

Glancing at subsequent pages in our original No. 1 issue, there are one or two design features that strike a familiar note. It appears, for example, that the Wolseley Company were in 1910 water-jacketing their induction manifold riser, a feature to which many cars are at present reverting. In this same issue there is a description of the 8 h.p. Rover car, a very sound little vehicle in its day. In the course of this description we read that the carburettor is the "standard Rover pattern." Of this mechanism, we note that "the holes in the rising and falling cone, and the strength of the spring are determined by means of experiment for each particular engine once and for all. Subsequently each car is tuned up by means of enlarging the size of the jet. Test drivers are provided with a small reamer, and each car is tested on a particular hill, the jet having to be enlarged until the car will climb the hill on direct drive under normal conditions."

The critical eye

We regret to note that in those days we regarded the Rover design as being "open to severe criticism on various technical grounds," yet admitted that it performed satisfactorily on the road, as "evidenced by the ability of the makers to obtain a fair price for the car for several years in succession." The principal part of the design with which we "feel inclined to find fault" was the "considerable difficulty of getting at any part of the engine, clutch and gear unit without dismantling the whole." Similarly, "a new part in the gear box could not be fitted without several hours of work."

Again, "to take up the big end, means not only opening the unit but necessitates the complete removal of the crankshaft and the detachment of one of the flywheels." We thought also that "the lubrication of the engine put rather too high a premium upon the memory and intelligence of the driver, and an automatic system would be an improvement." . . . "The pot which has to be filled at regular intervals is no doubt less trouble than a gravity drip feed, and less likely to be abused." It is interesting in view of these comments to consider the position of the Rover Company to-day, more particularly, their recent great achievement with the turbine car.

On other pages of the No. 1 issue, there was an article entitled "Automatic Engine Starting," and we regret to say that we then regarded "such a device as essentially a luxury, perhaps in the same sense that a paid driver is." We imagine that "it is on this account that the attention given to the solution of the problem has not been so serious as that given to other problems such as increased efficiency and economy." We admit that "those manufacturers who fit seat starters or self starters to their cars find that they meet with enthusiastic approval." We therefore safely conclude that "there can be little doubt that for a reliable, moderately priced article of the kind, there would be considerable demand."

On the subject of road testing chassis and that rascally little reamer, it is interesting to note that one of our contributors is already criticizing the quality of the final test. He agrees that in many small works this system works extremely well, because some person in a responsible position makes the final test personally, but he complains that in the larger organizations the final testing is in the hands of "men whose only interest is a wage of a few pence per hour."

Production work

We were already alive to the importance of production, and regarding manufacturing methods there is in issue No. 1 a comprehensive illustrated survey of grinding machinery, the accent being on the saving in cost of production rather than accuracy. The accuracy of the grinding process is, however, fully appreciated, and reference is made to the close sizing and finishing of the work. In concluding the article, which is illustrated by 13 half-tones and one line illustration, there is a reference to a third grinding machine. The purpose of this machine is to grind the valves into their seatings, not to finish the valve itself.

The author states "at first sight it might seem to be superfluous to have a machine for such a purpose, for the grinding-in of valves is usually looked upon as quite a trifling matter. If, however, we can consider the case of a factory turning out say 20 cars a week, which means 80 cylinders and 160 valves, the money saving effected by this machine is worth consideration." Although the machine is fully described in the utmost detail, it is quite obvious from the illustration that it is just an ordinary vertical drill applied to this purpose.

In 1950 there are not so very many automobile engineers who remember these interesting, anxious, enthusiastic days. Later generations of engineers and designers have, however, carried on the work and more than fulfilled the dreams of the pioneers. "Vintagents" to the contrary notwithstanding, the 1950 car constitutes indeed a realization of that "ever-improving automobile" envisaged in our original leading article.

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THE B.M.M.O. BUS CHASSIS

A Detailed Description of the S.10 Type Chassis

OPERATING a fleet of about 1,800 buses, both single- and double-deckers, in a large area with Birmingham as a centre, the Birmingham and Midland Omnibus Co. Ltd., have, for many years, designed and built their own vehicles. It should be pointed out that the Birmingham concern do not supply vehicles to other operators or do any export business. This fact explains some features in the chassis about to be described, such as the engine size, for example, which is of 8 litres and gives some 100 h.p. as compared with say 9½ litres and 125 h.p. now becoming accepted practice where export markets, frequently involving the use of much larger and heavier bodywork, are concerned. Again, the firm find that, on their routes, a radiator cooling fan is not necessary. The elimination of this clearly results in a saving of some horse-power in the average engine output, with a measurable reduction in fuel consumption.

Since the firm is its own and only customer, it is favourably situated in respect of development and experimental work. Comparative trials of alternative and new features are continually in progress and these can often be done as large-scale tests. Batches of 100 or more vehicles can be put into service with some modification which has given promise on a single example. Research and development on these lines is difficult in the case of a firm with many customers. Modifications are likely to inconvenience the various operators, and also complicate the spare parts and service organization.

Again, there are certain alternative design details between which there does not appear to be much to choose. A manufacturer with many customers may be perfectly right in adhering to some particular practice, even though slight improvement were possible by a change. In the Birmingham and Midland organization great importance is attached to research and experimental work, although in some cases the results may be only applicable in full force to the Midland Red operating conditions.

SPECIFICATION.

ENGINE. 8-litre horizontal, six-cylinder, direct injection with toroidal combustion chambers in pistons. 4.45in. bore by 5.25in. stroke. 102 b.h.p. at 1,700 r.p.m.

TRANSMISSION. Single-disc clutch, Layrub shaft to separate four-speed gearbox. Top, 3rd and 2nd speeds constant mesh, ground helical gears. 1st speed sliding. Ratios 1, 1.67, 2.743 and 4.55 to 1. Mechanics propeller shaft to back axle.

REAR AXLE. Fully-floating, overhead worm, offset. 7¾in. centres. Ratio 5.5 to 1.

SUSPENSION. Leaf springs with positively lubricated shackles.

BRAKES. Lockheed full-flow hydraulic servo on all four wheels by pedal. Hand brake on rear wheels only.

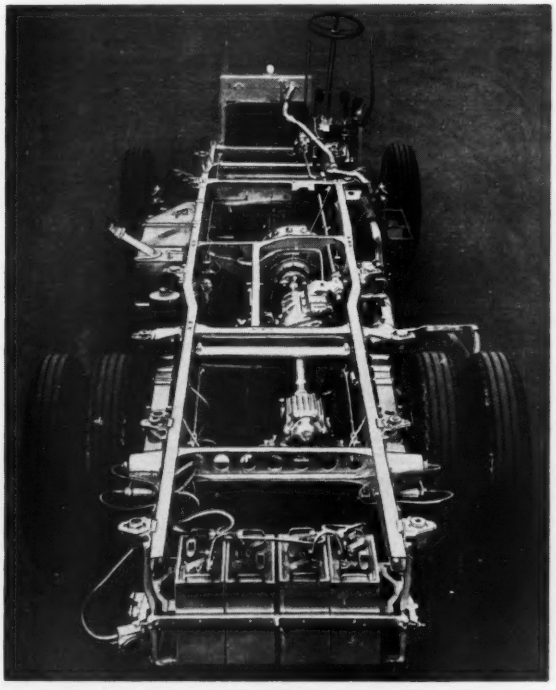
TYRES. 10.00 x 20, singles on front, twins at rear.

DIMENSIONS. Wheelbase 16ft. Chassis intended for 40-seater body 8ft. wide.

The S.10 chassis

Designed to take an all-metal steel constructed single-deck body seating 40 passengers, the chassis is extended to allow of forward entrance. The wheelbase is 16ft 0in and the laden frame height about 2ft 9½in. The body is a separate structure, the floor being 4⅜in above the frame rails, except at the front, where it is sloped down 2¼in to the front entrance.

The off-side frame rail is raised 2⅞in for part of its length to clear the underfloor six-cylinder C.I. engine, which drives, through a disc clutch, a separate four-speed gearbox, with dog-clutch engagement of the three higher gears. The floor being fairly high, an underhung worm is not necessary and the rear springs are above the axle, the frame being inswept 2⅛in each side just in front of the rear spring front brackets. The unladen weight of the complete vehicle is 6 tons 13 cwt 2 qr, fuel consumption in service is about 12 miles per gallon.



The B.M.M.O. bus chassis.

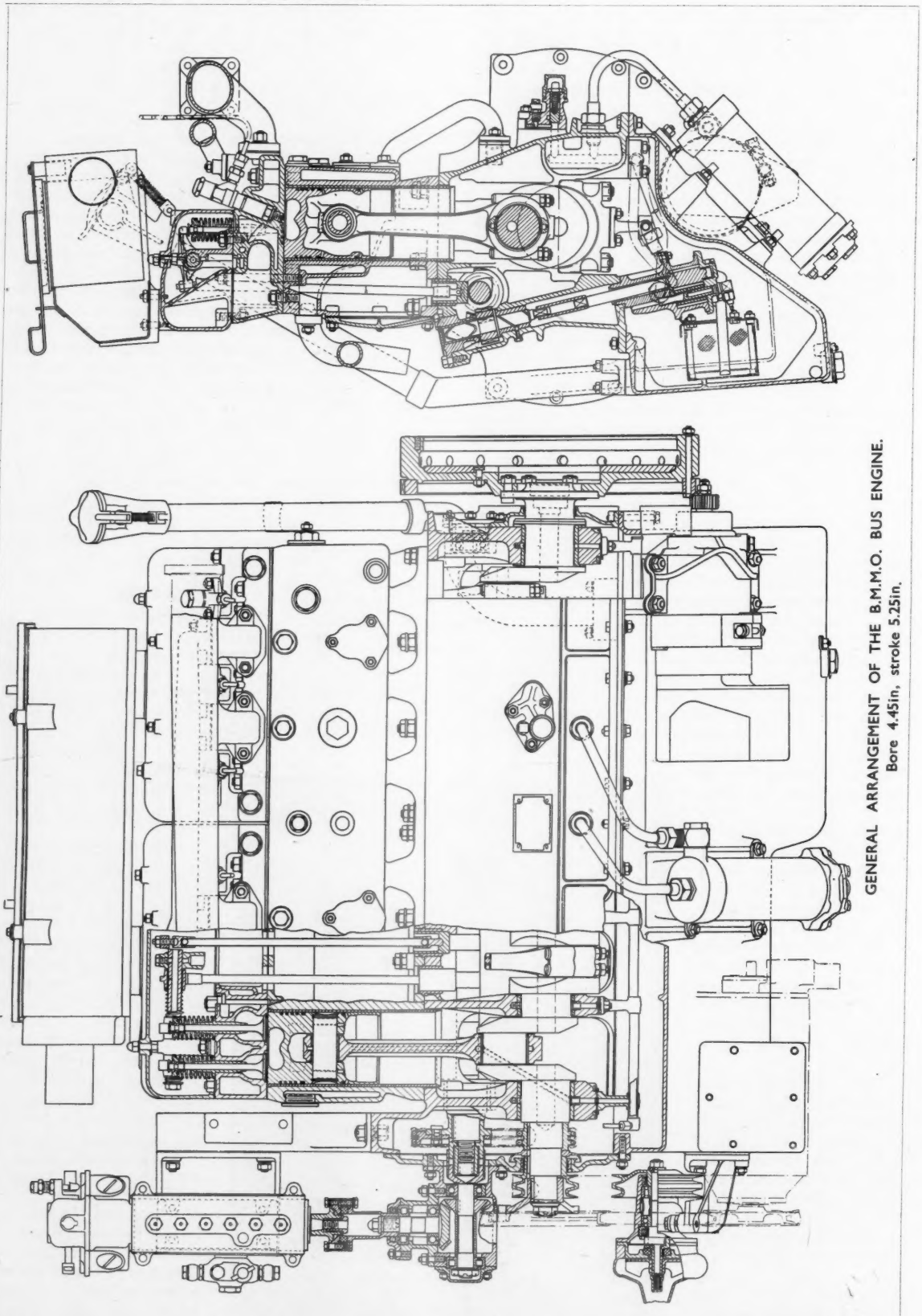
Engine

With a bore of 4·45in (113 mm.) and a stroke of 5·25in (133·3 mm.) the swept volume is 489·9 cubic inches (8,028 c.c.). The compression-ratio is 17 to 1 and the maximum brake horse-power 102 at 1,700 r.p.m. Maximum torque is 345 lb/ft at 1,200 r.p.m. at which speed the b.m.e.p. is 105 lb per sq in. Full load fuel consumption is 0·35 pints per b.h.p./hour.

Inclined upwards at an angle of 5 deg the cast-iron cylinder block is secured to the cast-iron crankcase by through-going bolts that hold the main bearing caps. Soft Brivadium liners are fitted with an interference fit of 3/1,000in, being honed after assembly. A number of engines are now being put into service with "push-fit" pre-honed liners and if found satisfactory in service this practice will be adopted.

An important point in connection with the relatively soft liners is the use of chromium-plated rings in the two top grooves of the piston. This combination, together with the use of two ordinary iron rings below the two slotted scraper rings, has given remarkably satisfactory re-

THE B.M.M.O. BUS CHASSIS (Continued)



GENERAL ARRANGEMENT OF THE B.M.M.O. BUS ENGINE.
Bore 4.45in, stroke 5.25in.

sults. The pistons are of low-expansion silicon-aluminium alloy and have fully-floating gudgeon pins 1½in. in diameter located by circlips. They are tin-plated to prevent "scuffing."

Cylinder heads

Two separate cast-iron cylinder heads carry inlet and exhaust valves both 1¾in. in throat diameter, located 29/64in below cylinder centre line. The diagonally-mounted injectors break through the head face on the axis of the toroidal combustion-chamber formed in the piston slightly above centre-line. Four-hole nozzles are used, with an included angle of 150 deg. They are dowelled in the holders to ensure correct positioning of the sprays. C.A.V. or Simms injection pumps are fitted, both having centrifugal governors and fuel supply by plunger-type lift pump.

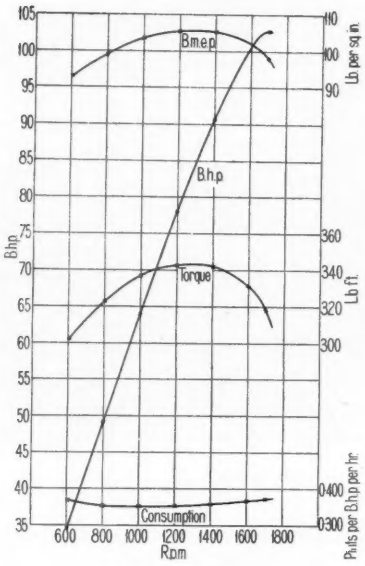
The one-piece exhaust manifold with upwardly-swept branches lies outside the main frame side member. Directly below it is the water outlet manifold, leaving a clear space for direct access to the injectors. The six injection pipes pass between these two, almost touching the exhaust manifold.

The air supply arrangements are of considerable interest. There is a gauze-protected port in the cab just above the driver's windscreen and this is connected by channels in the body structure and by an external pipe to a large box bolted to the two aluminium valve-rocker covers.

This box, which has a quickly-detachable cover, contains, in a tray filled with oil, a felt mat which in its turn contacts the lower edges of a number of staggered vertical baffles built up of lamp-wick on framing. The air passes through the labyrinth thus formed and then by two ports into two manifolds cast integral with the rocker covers.

Crankcase and crankshaft

The cast-iron crankcase is cast separately from the cylinder block and the sump joint face is well below crankshaft centre line. The main bearings are 3⅛in. in diameter by 1 7/16in overall length except the front, centre and rear bearings, which are 2½in long. The steel bearing shells are dowelled in position, those in the caps being lined with lead-bronze



Performance curves.

while the crankcase shells are lined with white-metal. The crankpins are 2¾in. in diameter by 1 1/16in overall length and have steel shells with lead-bronze lining. The standard crankshaft is made of Tocco-Hardened 1 per cent chrome-molybdenum steel and the big-end shells are unflanged, the rods themselves giving end-location. Tests of nitrided 3 per cent chrome-molybdenum shafts are being made in a considerable number of engines, but service experience has proved that the nitrided shaft has nothing to offer from a life point of view when compared with the Tocco shaft.

The advantage of the Tocco-hardened shaft is that it can be reground twice, giving a service life of about 350,000 miles before scrapping, as compared with about 200,000 miles for the nitrided shaft, which cannot conveniently be reground.

The case-hardened camshaft runs in steel-backed white-metal bushes and is driven by a duplex roller chain provided with a tensioning idler. The crankshaft and idler sprockets are made of 0.6 per cent carbon steel while the camshaft chain-wheel is of close-grained cast iron. End-location of the camshaft is by a ball journal bearing at the rear. The hub of the chain-wheel is extended forward and has dogs that engage with dogs on a subsidiary shaft. This is carried on one ball journal bearing and one roller bearing, giving end location, and having keyed to it a bevel wheel. This drives a similarly-mounted wheel integral with a short spindle driving the fuel pump which is outside the main frame on the assembled chassis.

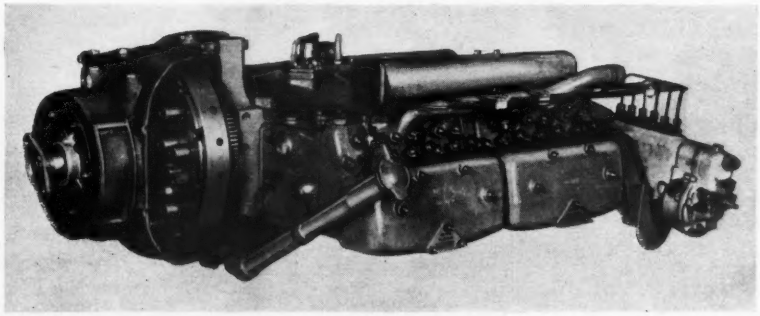
Cooling

Twin V-belts from a small pulley on the front of the crankshaft drive the spindle of the centrifugal water pump at about three-quarter engine speed. This spindle is mounted on one ball and one roller bearing. The spindle drives, by dogs, the flanged pump impeller centre, which is slidably carried by a spigot extension and is forced against a carbon ring by the axial compression spring.

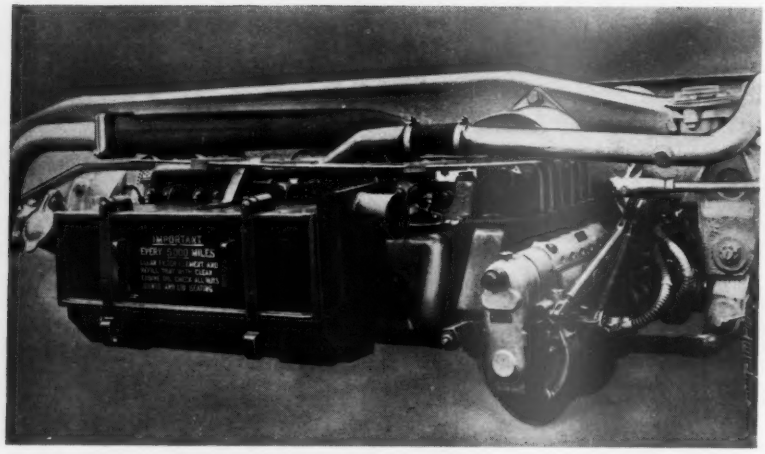
The tubular radiator is mounted in the front of the chassis, the total capacity of the system being 10 gallons. The header tank is deeper than usual and allows for considerable loss of water before the tube-plate is uncovered. No fan is fitted, but the tube stack is not blanked off by the body floor, so that the air stream caused by the vehicle movement has a remarkably free flow through the radiator. There is much less obstruction than is the case with a radiator in front of a conventional bonnet filled with engine and accessories.

Lubrication

It has been found possible to provide a sump with a capacity of 7 gallons while retaining a ground clearance of 12in and without resorting to the "dry sump" system. Driven by spiral gears from the camshaft, the gear-wheel pump draws oil through a large gauze strainer and, at 1,700 r.p.m., delivers 5¼ gallons a minute at 50 lb per sq in. The piston-type relief valve is mounted on the crankcase, readily accessible through a trap.



The six-cylinder underfloor engine.



Injection pump mounting and air-cleaner.

After passing through a large Purolator filter of the wire-wound type (spacing 0.0015-in) the oil passes to a tubular gallery carried by the main bearing caps and by the usual oilways to the big-end bearings. A separate feed is provided for the valve rocker gear. A snap-action oil filler cap is carried on a long tubular extension with the dip stick inside it.

As previously mentioned, the oil capacity is seven gallons, and in service the sump is drained every 7,000 miles. Normal mineral oil of low price is successfully used without any detergents, and is moreover reclaimed for re-use in the engines. This process includes a trisodium phosphate wash, centrifuging, boiling in vacuo at 600 deg. Fahrenheit and final filtering.

Clutch

Of the firm's own design, the single-disc clutch has an external diameter of 16in, the inside diameter of the facing being 6½in. A total lining surface of 336 sq in is provided and the spring pressure is 2,010 lb. With a view to an increase in life between overhauls, a twin disc clutch is now coming into service. This has discs 13¼in outside diameter and 6½in bore, giving a total area of 412 sq in.

The actuating mechanism is particularly robust. Three withdrawal levers are used, with press-fitted case-hardened fulcrum pins, turning on needle rollers. The outer races of these are housed in two blind ended blocks studded to the clutch cover. Felt grease retainers are fitted, there being a nipple to each block. Side-location of the levers is effected by arranging the ends of the fulcrum pins to contact the blind ends of the bearing blocks. The levers run tangentially to the withdrawal ring, to which they are attached by loosely-fitting pin joints having case-hardened bushes.

The clutch stop arrangement is usual, the annular facing being riveted to a flange within the withdrawal ring and slidably fitted to the clutch shaft. An axial compression spring forces the flange against a shoulder and serves to limit the braking effort.

The whole assembly is housed in an aluminium bell with a malleable-iron centre taking the clutch shaft support bearing and extended to carry the sliding withdrawal bearing unit. Connection with the gearbox is by a shaft fitted with Layrub joints.

Engine mounting

This has received careful study in view of the underfloor positioning of the power unit. The front of the crankcase is carried by two very large bonded rubber bushes with a considerable radial thickness of rubber, the durometer hardness being 40. One bush is slightly to the right of the crankshaft line, approximately on the mass-centre of the engine, and the other, which is axially nearly in line with it, is just outside the main frame side member and takes a major part in limiting angular movement.

At the rear, the back of the aluminium clutch housing has three shallow recesses, one on each side and one at the bottom. These take segmental rubber pads, the outer circumferences of which are clamped in a banjo pressed steel cross-member, the lower part of which is detachable when dropping the power unit out of the frame.

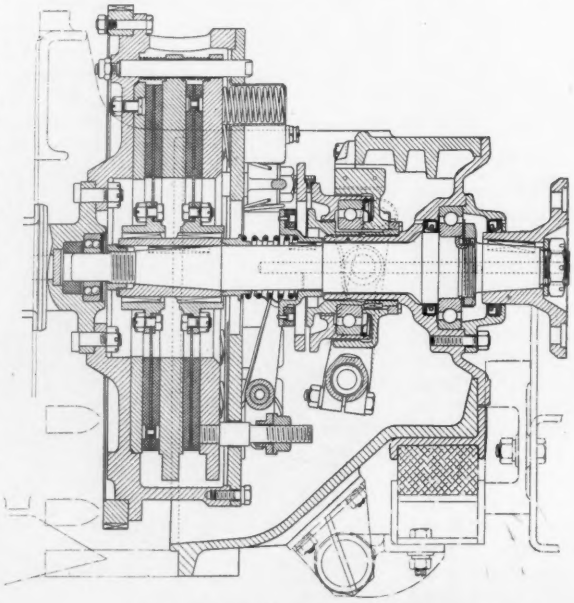
As the gearbox is a separate unit, the engine mass it not subject to the push-and-pull action of the propeller-shaft sliding joint, but it has been found desirable to fit a Ferodo-faced plate, bearing on the rear face of the banjo, to prevent end movement due to the pull on the clutch withdrawal rod.

The silencer is outside the side-member close behind the manifold. A rigid steel pipe connects the two, but, to allow for the vertical oscillation of the cylinder heads and manifold, the silencer shell is hung by clips from the ends of a balance beam free to rock on a pin. The tail pipe is rigidly attached to the body but connected to the silencer by flexible metallic tubing.

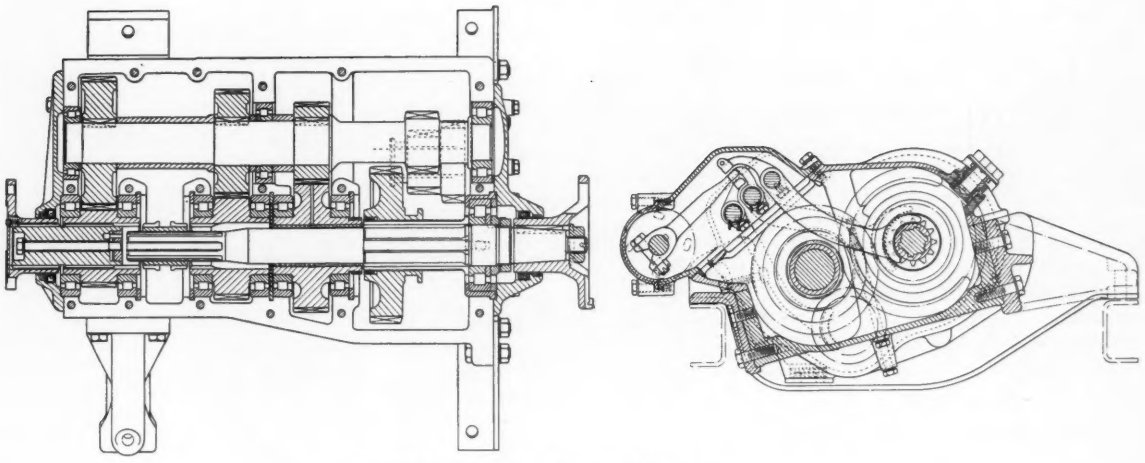
Gearbox

This unit has a split cast-iron casing in which the constant-mesh pinion and third and second-speed driven wheels are mounted on pairs of roller bearings giving end-location by opposed flanges. Separate inner races are employed and the wheels have helical teeth, ground after hardening. The layshaft is carried on three roller bearings, of which the two end ones give axial location.

First speed and reverse are given



Arrangement of twin disc clutch.



General arrangement of four-speed gearbox.

by a straight-toothed wheel sliding to the rear to mesh with first speed and having internal dog teeth which engage second speed when moved forward. Reverse is engaged by sliding a two-piece idler.

The driven shaft is carried at the rear by a roller bearing with flanges giving end location in both directions. It is additionally supported in a bush pressed into the bore of the second-speed driven wheel. To the overhanging front end is splined a small double-ended dog clutch engaging internal teeth to give direct drive and third speed.

A separate aluminium casting carries the striking fork assembly, which is orthodox in layout, though the more usual caliper locking might be expected instead of the ball interlock actually employed. The selector finger is clamped to the customary sliding and rotating shaft coupled to the operating shaft below the change-speed lever by a shaft, with an intermediate support, made in three pieces for ease of assembly.

The ratios are :—Top 1 : 1, third 1·67 : 1, second 2·743 : 1, first 4·55 : 1 and reverse 5·88 : 1.

Four-point mounting of the gearbox is provided, rubber bushes being used throughout, the two at the rear arranged with their axes horizontal, to take torque reaction. The two at the front, not having to perform this duty, can be set with their axes vertical, giving a better cushioning effect.

Back axle

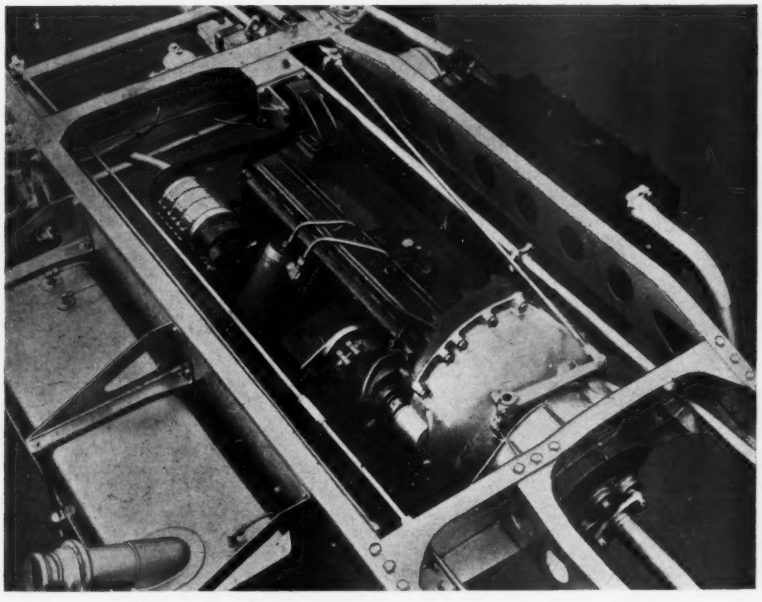
Giving a ratio of 5·5 to 1 the back axle is of normal design, the $7\frac{3}{4}$in centre worm being carried in an elektron housing. The floor height of the vehicle allows the worm to be mounted on top, which involves a downward run of the Mechanics-jointed propeller shaft to the gearbox. To reduce angularity the axle is turned slightly downwards.

The wormshaft runs in parallel roller bearings with a duplex ball thrust, fed with oil by trough and drilled passage into the cap. The differential is also carried on parallel roller bearings with ball thrust washers adjusted by shims. The bearing caps are spigoted into the drop-forged axle housing for additional support.

The Company is working on the design of a hypoid axle. With the relatively high gearing associated with C.I. engines, it should be easy to get adequate strength in the bevel pinion, while the transmission efficiency should be higher, the propeller-shaft layout more satisfactory, and a lower floor-line available if required.

The rear hubs are mounted on taper roller bearings, the adjusting ring-nuts for which are extended within the race bore and butt on shims seating on a shoulder on the axle sleeve. This is a simpler means of predetermining the wheel-bearing adjustment than the long tubular distance piece sometimes used.

A feature of the brake mechanism, is the mounting of the camshafts on parallel rollers, well spaced and protected by felt washers at both ends. The drums are 17in. in diameter and the linings only $\frac{7}{16}$in thick, though $6\frac{3}{4}$in wide. It must be understood that the chassis is not intended



Plan view showing engine installation and mounting.

THE B.M.M.O. BUS CHASSIS (Continued)

for export, or tropical conditions, where a hot brake drum near to the wheel is undesirable. Even so, it would seem that a smaller, thicker, drum together with a thick lining is a better compromise.

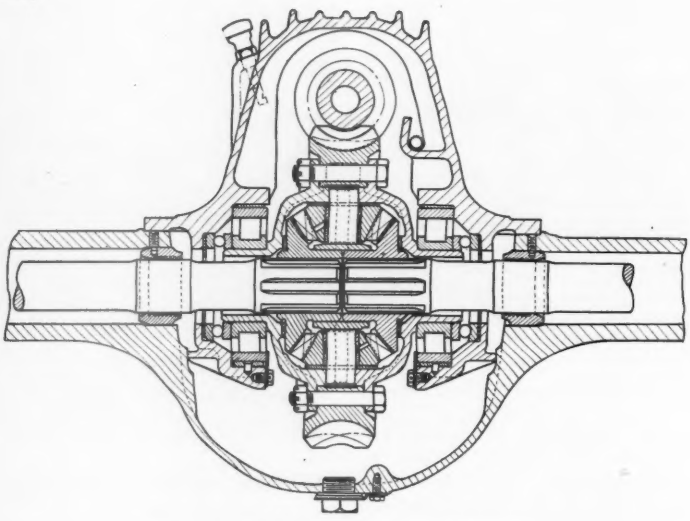
The Company have a great deal of experience with all types of brake lining and are coming to the conclusion that moulded linings give better life than woven ones. Quite remarkable life was evidenced by some American moulded facings, which had a coefficient of friction comparable with standard types of British linings.

Front axle

Having a track at wheel centre line of 6ft 11⅞in and spring centres of 3ft 4½in, 45 deg of inside lock is provided. At the top the swivel pins have taper roller bearings taking both journal and end load, with plain bushes at the bottom. These are integral with the bottom cover flange and can be removed and replaced without dismantling the swivel assembly or removing the swivel pins from the axle bed.

Inclination of the pivots is 5½ deg and the camber 1 deg, the offset at ground contact point being 1¼in. The hubs run on taper roller bearings and the hub grease seal comprises a felt washer recessed into the swivel stamping compressed by a ring on the hub having an annular ridge biting well into the felt.

The brake drums are 17in. in diameter and the shoes 3½in wide with $\frac{7}{16}$in linings. The cam spindle is mounted on a Hyatt type flexible roller bearing, protected by felt seals both inside and outside. The bearing seems a little short, and rather removed from the centre of the brake cam in view of the unbalanced loads set up by the operation of rigid (i.e., non-floating) shoes. Brake adjustment is of the scissors type and the Lockheed operating cylinders are 1¼in. in diameter and work with a cam-to-piston ratio of 5·44 to 1.

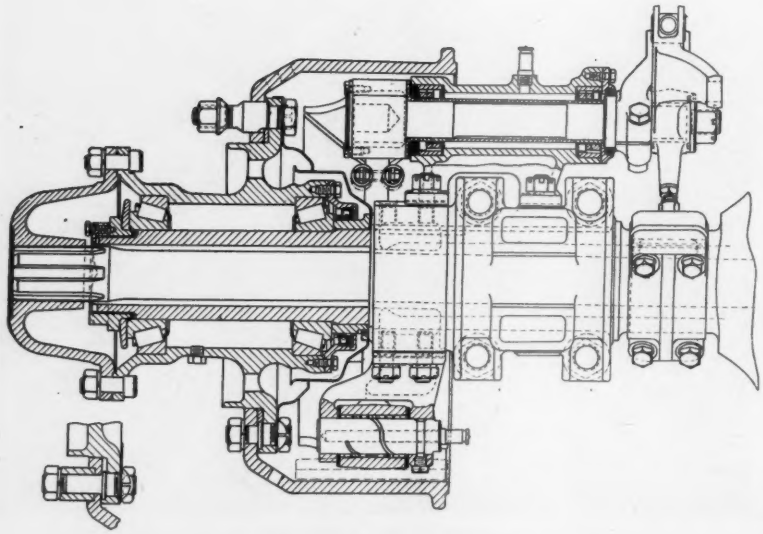


Arrangement of worm drive and differential.

Brake operation

A cross-shaft in the front of the chassis and a longitudinal rod, link the servo of the Lockheed continuous flow system to the pedal. The servo is on the near-side of the chassis attached to the cross-member forming the rear engine mounting. Pipes from the servo lead to four cylinders mounted directly on the axles, those at the rear being 1½in. in diameter and working with a piston-to-cam ratio of 5.3 to 1. The radial piston brake pump is belt-driven from the gearbox and, in this system, no accumulator is employed.

Brake power is more than ample, a retardation of 74 per cent. being obtained at 20 m.p.h. and 69 per cent at 30 m.p.h. The handbrake operates through rodwork on the right-hand side of the chassis to a large diameter tubular cross-shaft just in front of the back axle. There is no compensation, and the overall ratio is 107 to 1.



Section through rear axle and hub.

Suspension

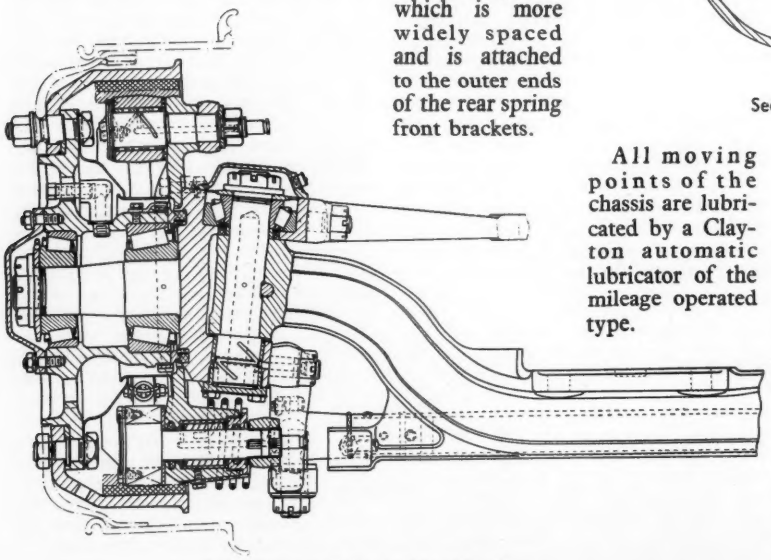
Considerable development work is in hand, including a coil-sprung independent front suspension and the Metalastik rubber suspension for the back axle. The standard chassis, however, still has leaf springs of normal design. The front springs are 3½in wide and 4ft centres, with a torque plate $\frac{1}{8}\frac{1}{2}$in thick, five leaves $\frac{1}{8}\frac{1}{2}$in thick and eight leaves $\frac{5}{16}$in thick. The rate is 1·52in per ton, normal load 3,842 lb and static deflection 2·6in.

At the rear the springs have the same width but with 5ft 6in centres. The torque plate is $\frac{7}{16}$in, five leaves $\frac{1}{4}$in thick and six leaves $\frac{7}{16}$in thick. The rate is 1·66in per ton, normal load 4,816 lb and static deflection 3·55in. All the springs have normal lubricated shackle-pins, piped to the 30-point Clayton-Dewandre lubricator which is belt-driven from the gearbox via a flexible shaft. They all have six clips each consisting of U-pieces riveted to the tips of the leaves and upstanding to take cross-bolts passing through tubular dis-

tance-pieces. Between these and the top leaf of the spring are rubber pads that are initially compressed on assembly by a clamp holding the distance-tube down while the cross-bolt is inserted.

Frame

The side members, of 3½% nickel steel are 8in deep by 2¼in wide, pressed from ¼in stock. All cross-members are of manganese steel. The overall length of the frame is 26ft 10in, the front overhang being 4ft 11in. and the rear overhang 5ft 11in. The width is 3ft 6¾in at the front, running parallel to a point just in front of the rear spring front anchorage, where it is inswept to 3ft 2⅛in, continuing at this dimension to the rear end. Upswept 2⅞in over the engine cylinder block, this portion of the offside member is reinforced by a boxing plate welded in position.



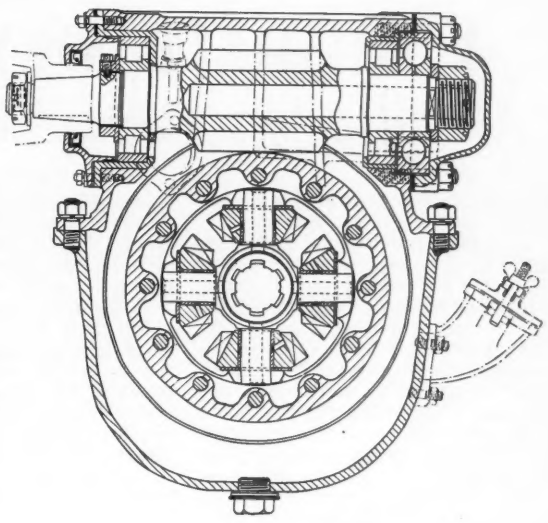
Arrangement of front hub and axle.

Cross members

A light channel pressing ties the tapered front ends of the side members in line with the Marles twin-roller steering box. There are channel members at each end of both front and rear springs, and an intermediate member forms the banjo support of the rear of the power unit. This member is connected to the rear spring front cross-member by a longitudinal channel nearly on the centre-line of the frame, which serves to support one side of the gearbox. The transmission line is offset 6 1/16in from chassis centre-line. All members have gusseted ends and are secured by fitting bolts, there being no riveting or welding except for the reinforcement already mentioned.

The body is mounted on one dozen rubber mountings of Silentbloc manufacture. These are bolted to the outside faces of the frame side members except for one pair, which is more widely spaced and is attached to the outer ends of the rear spring front brackets.

All moving points of the chassis are lubricated by a Clayton automatic lubricator of the mileage operated type.

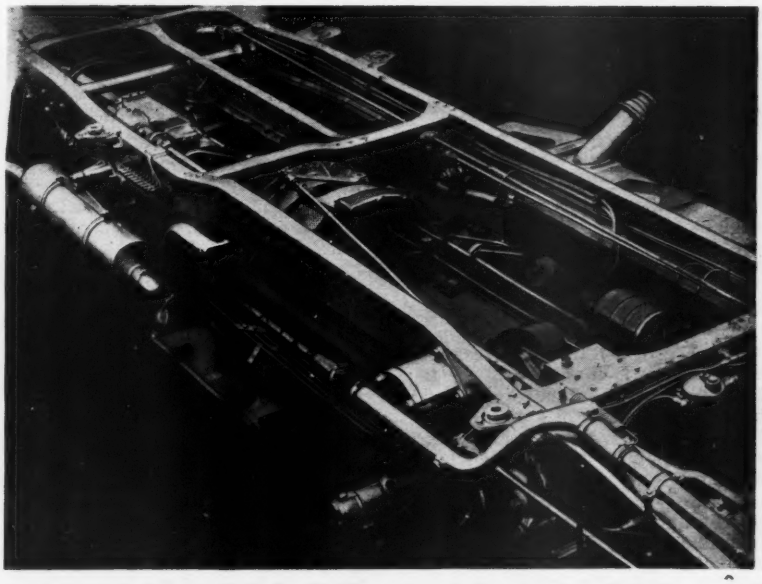


Section through worm drive.

Electrical equipment

A 24-volt lead-acid battery is mounted in a cradle spanning the extreme rear of the frame, the retaining cross-bar being secured by swing-bolts permitting instant removal of the battery. Capacity is 185 ampere-hours. The 800-watt non-ventilated dynamo is belt-driven from the front end of the crankshaft while the axial starter is mounted on the engine sump.

The speedometer counts as part of the electrical equipment, being of B.T.H. manufacture and comprising a belt-driven generator connected to a voltmeter calibrated in miles per hour.



Engine and transmission installation.

A REMOTE DIAL FLOWMETER

Details of a New Instrument by B. Dixon-Bate, Ltd.

WHILE the flowmeter is a standard piece of equipment on the test bench it is probable that most development engineers have wished for a device of similar nature which could be applied to a vehicle. In this way, continuous readings of fuel consumption could be obtained on the road, whereby immediate information would be evident on the effect of adjustments. This would apply not only to the power unit itself and its accessories but also to the vehicle and its running gear.

Such a portable flowmeter has been developed by B. Dixon-Bate, Ltd., Bridge Works, Chester. It is self-contained, its measuring unit being inserted in the fuel line of the engine. Its indicating component is a normal instrument dial that can be placed at any convenient point since it is electrically actuated.

Normal flowmeter principles are employed, that is, an open-topped vertical tube having a suitable size of jet near its base is fed with fuel at slight pressure through an orifice in its foot. The fuel rises within the tube and also passes out through the jet to the engine and according to the rate of flow through the jet a column is supported in the tube. Its height is accurately related to the rate of flow through the jet, so providing a measure of the rate of flow.

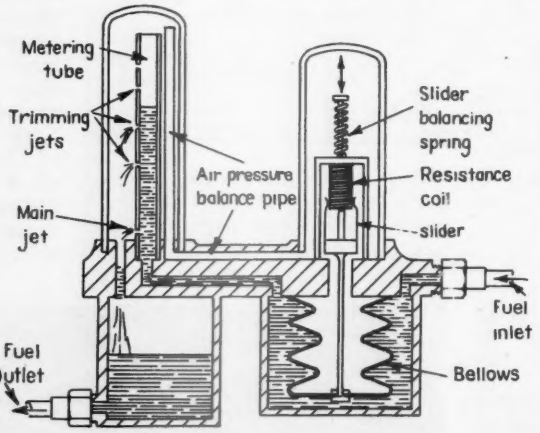
In the Dixon-Bate flowmeter, the varying height of the column in the metering tube is not measured as a dimension but is the means of providing a varying pressure due to increase or decrease of head. This small pressure variation is applied to a bellows unit which moves the slider of a rheostat unit. In this way it actuates an indicating instrument of the fuel tank contents gauge type having two opposing field coils.

The measuring unit of the device is a hermetically sealed container embodying the flowmeter and the pressure-sensitive variable resistance assembly. It is connected in the fuel line between the lift pump and the carburettor in the case of a petrol engine. For oil engines it must be located in a gravity supply to the injection pump since the flow between the usual lift pump and the common gallery of the injection pump is not necessarily equal in volume to the fuel actually injected into the cylinders. On many oil-engined vehicles the lift pump discharges into a small constant-level gravity tank or into a filter unit with a by-pass valve which gives the same effect. The flowmeter would be connected between such a point and the intake union on the injection pump.

Fuel flowing through the flowmeter passes first into the metering tube, issues from the measuring jet and runs into an outlet chamber from which it is led to the engine. There is also a second chamber containing a bellows which is resistant to the chemical effect of the fuel. The bellows is completely submerged and would be compressed were it not internally pressure balanced through an air pipe connected to the metering tube chamber.

As fuel rises in the metering tube and collects in the outlet chamber the contained air is compressed to the pressure of the fuel supply system. By way of the balance pipe this pressure is conveyed to the interior of the bellows which is thus maintained in equilibrium irrespective of the fuel line pressure.

It is free to respond to any pressure unbalance which may be superimposed on the steady delivery pressure. Such an unbalance is created by the head of fuel in the metering tube.



Attached to the free end of the bellows is a vertical rod carrying a pair of balanced slider contacts which engage with a fixed resistance coil. The rod is supported by a compression spring which balances its weight and that of the bellows. The bellows unit is extremely sensitive so that it will respond to the exceedingly small pressure changes due to the varying head of fuel in the metering tube which is only some five inches long and about ½in bore. Electrical connection to the slider rod fitted to the synthetic rubber bellows is by a pig-tail at its lower end. Compactness has been attained by reducing the length of the metering tube by providing additional trimming jets above the main jet. These can be so positioned and dimensioned as to provide a linear reading on the indicating gauge if this is required. Calibration is in gallons per hour or any other quantity/time relationship that may be required.

The casing of the metering unit is a substantial light alloy casting with a brass cover plate which embodies the metering and resistance chambers. It is comparable in size say to a large magneto or fuel injection pump. It can, therefore, be rigidly fitted directly on to the engine itself or on the test bed, being connected in the fuel line at the appropriate point. A simple electric circuit only is needed for the indicating instrument which can be remotely placed wherever it is convenient to the operator.

Readings with no more than one per cent error are possible and while it may not be true to say that the indication is dead beat, the lag is so small that the pointer very closely follows throttle movement or load variation. When fitted to a vehicle the indication appears to respond at least as quickly as the average speedometer and the instantaneous rate of fuel consumption can be checked with every change of driving conditions. The most economical speed for the car is readily found and the effect of small alterations in jet sizes, fuel delivery, or ignition timing are clearly shown. The comparative consumptions of full throttle top gear, or small throttle climbing on lower ratios, can be readily seen.

Naturally the compactness of the equipment and its continuous reading on a panel-type instrument have suggested the possibility of a miles per gallon meter and experimentally this has been examined. By using a speedometer with a moving dial and having an inversely calibrated log-scale for m.p.g. the desired result is obtained. The normally scaled moving dial passing a fixed pointer indicates the speed. A fixed scale is swept by the flowmeter pointer to show consumption in gallons per hour while the concentric log-scale on the moving dial passing under the same pointer gives the consumption in m.p.g. With a speedometer of the electric type, the flowmeter circuit is energized from the speedometer generator unit.

Apart from its application to fuel consumption readings the Dixon-Bate meter is applicable to any purpose where a rate of flow meter is required.

SPECIAL PURPOSE MACHINES

A Review of Some Recent Developments at Vauxhall Motors, Ltd.

THERE are two outstanding facts about the new machining line installed by Vauxhall Motors Ltd. for the production of the improved cylinder head for Bedford engines. They are (a) the great reduction in the total time cycle and the consequent increase in the total production from the line, and (b) the even greater increase in productivity per man-hour. These results may be ascribed to two factors, clever process planning and the use of machines specially designed for specific purposes.

A general description of this cylinder head line was published in *The Automobile Engineer* for May, 1950. As several of the machines afford such good examples of the important part that the machine tool industry of this country can play in the drive for greater productivity, fuller description is merited. Although these notes deal only with the machines themselves, it should be stressed that the results obtained have been made possible only through the closest possible co-operation between the machine tool builders and the planning engineers of Vauxhall Motors, Ltd. For example, the work-holding fixtures are of fundamental importance. They are the responsibility of Vauxhall Motors, Ltd.

The machines described in the following notes have been developed by James Archdale and Co., Ltd., Ledsam St., Birmingham 16, and John Lund Ltd., Eastburn Works, Cross Hills, Keighley, Yorkshire. An important point about the machines is that while each one has been developed for one specific operation, they all incorporate either standard units or modified standard units. This brings several advantages. Of these, probably the most important is that the use of standard units reduces the time necessary to develop the machine. Secondly, it reduces the capital cost. Finally, when the inevitable design changes are made, the complete machine does not necessarily become obsolete. On the contrary, substitution of one or more new units will probably allow the use of the machine on the newly designed component.

Special vertical millers

In planning the process sequence for the improved Bedford cylinder head, it was decided to finish machine the rocker face in one pass on a milling machine and to semi-finish the joint face in one pass on a second milling machine. This entailed the removal of approx. $\frac{3}{16}$in of metal from the rocker face and approx. $\frac{1}{8}$in of metal from the joint face. Not only were these heavy cuts to be taken, but in addition a high feed rate was necessary to keep the time cycle for these machines in balance with the time cycles for other machines in the line. Further, a good quality of surface finish and parallelism within close limits between the two faces had to be maintained.

There is no standard machine of appropriate size with sufficient power and rigidity to give the necessary rate of metal removal and the specified standard of finish, so James Archdale and Co., Ltd., have produced three special vertical millers, one for finish machining the rocker joint face in one pass, one for semi-finish machining the joint face, and one for finish machining the joint face. These machines are identical except that the two machines working on the rough casting use 40 h.p. motors for the spindle drive, while the machine for finishing the joint face has a 15 h.p. spindle drive motor.

One of these machines is illustrated in Fig. 1. The bed unit is a modified form of that used for the standard Archdale unit construction milling machine. To suit the height of the conveyor track, the table height has been kept low. Individual motors are employed for the feed and the quick-power traverse table drives. A $7\frac{1}{2}$ h.p. motor supplies power for the feed drive. The feed mechanism is based on the Archdale standard mechanism, but because of the very heavy duty for which these machines are to be used, a larger diameter screw and a special feed box are fitted. Special provision is made for eliminating backlash between the table screw and the nut. The available feed range is from 20in to 60in per minute, with changes effected by means of pick-off gears. Quick power traverse is at the rate of 300in per minute. This drive is taken from a 5 h.p. motor. The traverses are controlled through adjustable dogs on the side of the table.

Despite the very high power that must be transmitted through the spindle and the necessity for complete freedom from chatter, it has been possible to incorporate a standard spindle and quill assembly in each of these machines. As the machine is to be used for only one purpose, the normal provision for a range of spindle speeds is unnecessary, but nevertheless, it has been possible to use a modified form of the standard drive arrangements. Actually, drive from the main motor is by multiple vee belts to a pair of large spur gears and thence to the spindle. Should experience show that some other spindle speed is desirable, modification may be easily effected by fitting different size vee rope pulleys. The large diameter spindle is carried in a quill that has 1in vertical adjustment to compensate for cutter wear. The quill is locked after adjustment.

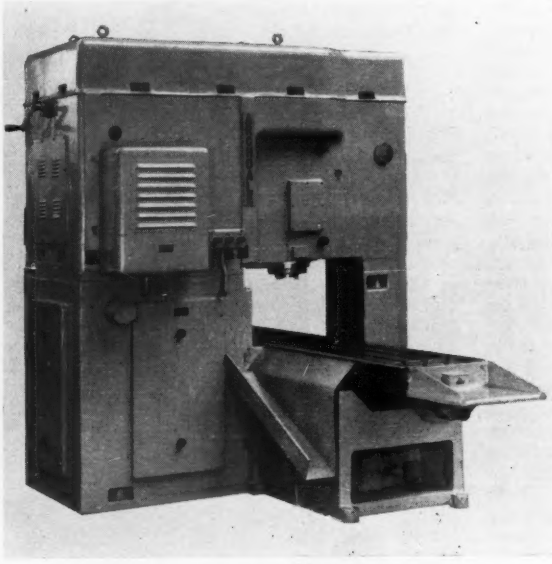


Fig. 1. Archdale vertical milling machine for machining cylinder head joint faces.

Not only is the spindle itself of substantial proportions, but it is also housed in a massive cross-rail type head which is supported by a main column and an auxiliary column. This construction gives complete freedom from spindle deflection under the heaviest cuts. The auxiliary column is on the operating side and is narrower than the main column which houses most of the electrical gear.

Under test conditions the machines with the 40 h.p. main drive have removed $\frac{3}{16}$in of metal of an average width of approximately 7in at a feed rate of 60in per minute. In production the machine has been used for removing $\frac{3}{16}$in of metal from a $7\frac{1}{2}$in face at a feed rate of 39in. The cutters are 12in diameter "Carbishear," with Ardalloy blades supplied by Alfred Herbert, Ltd.

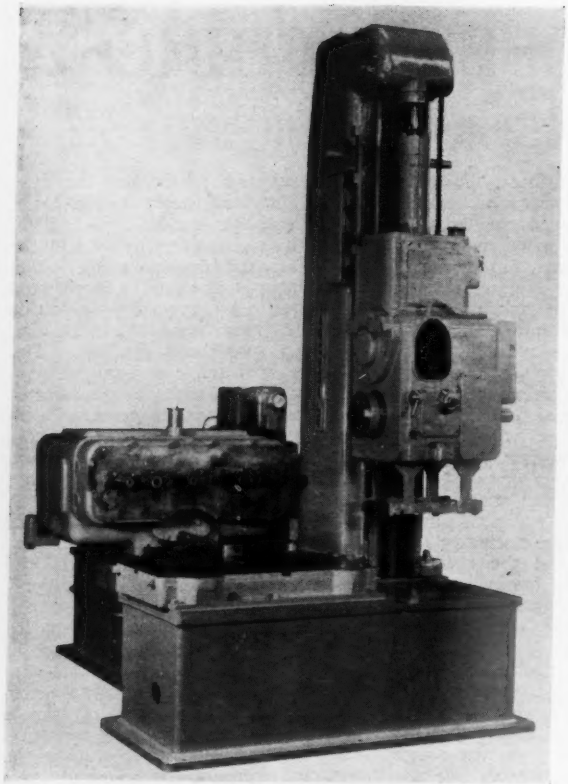


Fig. 2. Archdale combined milling and drilling machine.

Combined milling and drilling machine

Cylinder heads, cylinder blocks and suchlike components frequently require comparatively minor milling operations on relatively small faces, which when dealt with singly call for the same loading and unloading times as much longer operations. However, as conventional milling practice is to move the work past the tool, these operations are not generally looked upon as suitable for combination with other forms of machining. None the less, in planning the process sequence for the Bedford cylinder head, Vauxhall Motors, Ltd., decided that for maximum efficiency a milling operation must be combined with a drilling operation. This meant, of course, that the work would be stationary and the milling cutter must traverse.

Once again the problem was submitted to James Archdale and Co., Ltd., and the machine shown in Fig. 2 was developed. This machine drills the manifold face and mills the front end of the head. A common base carries a standard Archdale limited adjustable multi-spindle drilling head, horizontally mounted, and the column and hydraulic fed unit of an Archdale hydraulic feed multi-drill. These units are completely standard, but the vertical head is fitted with a special plunge cut milling head instead of the normal multi-drill head.

The drilling head carries eight twist drills and three core drills and the drive arrangements are such that all the drills work at the correct cutting speed. Hydraulic feed motion is employed. It gives fast approach at 230 inches per minute, any desired feed rate in the range 0 to 9in per minute, and fast return at 330in per minute. As is well-known, any one of several automatic cycles may be employed with this type of head. In this case the cycle is: fast approach, feed rate to a pre-determined depth, dwell for a predetermined period, fast return and stop. The hydraulic feed unit for traversing the special plunge cut milling head is the same type and capacity as that used in the horizontally-mounted drilling head.

These two units are in effect two separate machines with the controls so grouped that in normal operation, one set of push buttons controls the operation of both machines. To facilitate tool setting, either machine can be run independently of the other.

Transfer machines

The two transfer machines designed and built by James Archdale & Co., Ltd., for this cylinder head line are excellent examples of the manner in which standard machines can be adapted for special purpose work. Each of the machines has four working stations, but one employs five heads and the other four. Standard Archdale Size 1 multi-drill heads with hydraulic feed are used throughout the machines. In fact, the only real departure from completely standard machines is that at some stations the heads are mounted on special angular columns to suit the angular disposition of the holes with which they have to deal. If individual machines were employed, the columns would be vertical and the necessary angularity would be obtained from the work-holding fixture. With transfer machines this is not possible and the column carrying the head must be set at the appropriate angle. A transfer machine is illustrated in Figs. 3 and 4.

It must be stressed that each head is a self-contained unit, with its controls grouped with those of the other heads. This is important. If the complete transfer machine were built as a single unit, the problems of alignment throughout and the pitching from one head to the next would be acute. However, as each of these transfer machines comprises completely separate units, and because of the special form of transfer mechanism developed by James Archdale & Co., Ltd., it is not necessary to have absolute accuracy of alignment and pitching. The permissible limits of inaccuracy are not great, but they are sufficiently wide to simplify the problems of installation and operation.

It is really the special form of transfer mechanism that makes possible successful operation even though there may be slight inaccuracies in alignment and pitching. In effect, the transfer rack comprises five transfer bars, with a flexible link between each pair of bars. This flexible link compensates for any slight inaccuracy in the alignment between successive work-holding fixtures. In addition, the distance between suc-

Fig. 3. Loading end of Archdale transfer machine. Standard multi-drill units are used at every station.

cessive heads is such that the travel of the transfer rack will not carry the bar from one station as far as the next station. It is, therefore, possible to have each bar accurately lined up with its own head and work-holding fixture, a comparatively simple matter.

So far as the pitch between one head and the next is concerned, the transfer mechanism is designed to make provision for slight discrepancies in the distance that the block must be carried to travel from one working station to the next. Every work-holding fixture incorporates a dead stop to limit the forward travel of the component. This step is independent of the transfer mechanism and is fixed in the correct relation to the several spindles in the head. Forward movement of the transfer bar is imparted to the component through a spring-loaded dog. Should the cylinder head contact the dead stop before the transfer rack has completed its forward travel, there is a backward pressure on the dog that causes the spring to compress. This relieves the dead stop of any undue pressure.

Provision is made for operating each head, independently, from its own push-button control panel, see Fig. 4. This independent operation is used only in setting-up. For productive operation, the controls for all the heads and for the transfer mechanism are grouped to give complete control of the machine cycle from a single quadrant-mounted lever. It is impossible for the machine operator to employ any other than the correct sequence. Briefly, the sequence is:—

(1) Move lever from the neutral to the index position. This starts the master motor and the transfer rack moves forward a predetermined distance. The rack is then automatically reversed to its original position and the lever returned to neutral.

(2) The lever is advanced to the location position. Movement of the component during (1) brings it up against the dead stop to give approximation location. Completely accurate location at each station is given by two plugs which register in reamed holes on the bottom face of the cylinder head. These plugs are actuated from the control box through a system of rods and bell cranks. Actuation of the plugs is effected manually and the operator can feel if there is any undue difficulty in bringing them into the locating holes. Not until all the plugs are home can the operator move the lever forward to the next station, which is

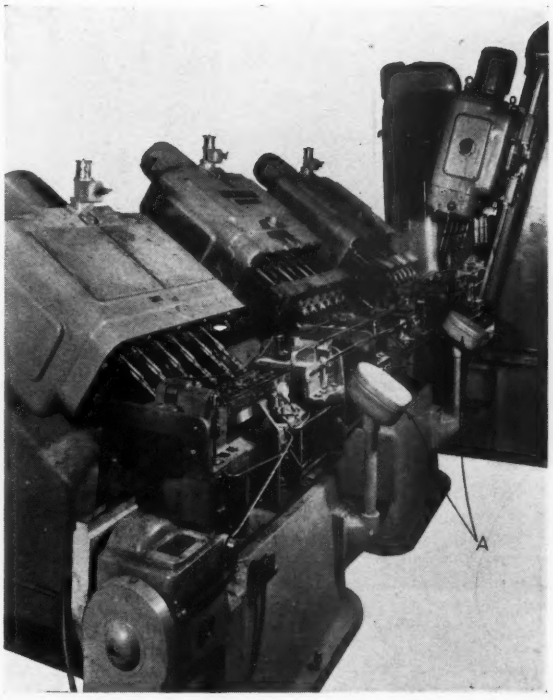


Fig. 4. Unloading end of transfer machine. Individual push-button control panels used only for setting up are shown at A.

(3) The clamping position. The clamps are all air-operated and movement of the lever to this position opens the air valves at the various stations. Each clamp has a micro-switch fitted and until the clamping is properly effected, the circuit is broken and the cycle cannot be continued.

(4) Further advance of the lever when clamping is completed starts the spindle and feed motors of all the heads. From this stage the cycle is completely automatic to give fast approach, feed rate, automatic timed dwell where necessary, and fast return. As each head returns to the starting position a warning lamp lights up. When all the heads have returned to the off position, the motors are cut out automatically and the lever returns to the neutral position ready for the initiation of another cycle.

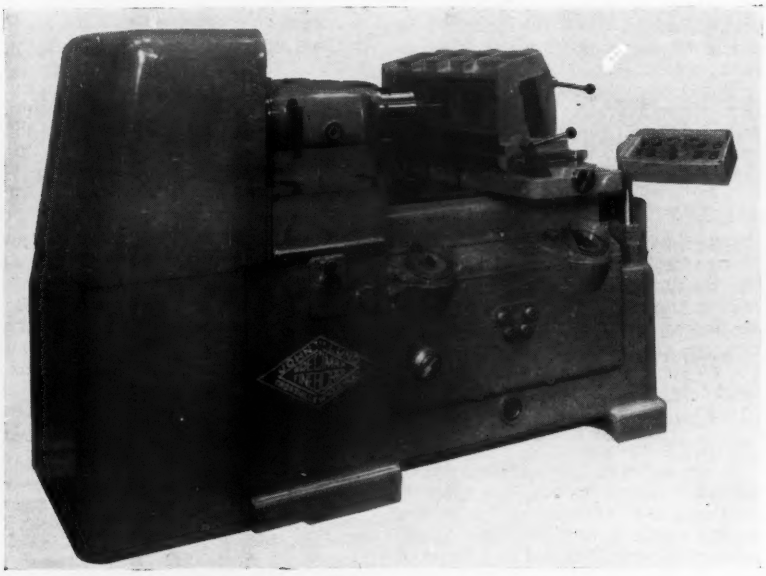


Fig. 5. Lund Precimax fine borer for valve elements in cylinder head.

Fine boring

Three special Precimax fine boring machines developed by John Lund, Ltd., Eastburn Works, Cross Hills, Keighley, Yorks, have been installed. Two of these are exactly similar and are used for machining the bore to take the valve guide, a counterbore below the valve seat and the valve seat itself at a single setting. One machine, see Fig. 5, is for inlet and the other for exhaust valve elements. In addition to the close dimensional accuracy that must be maintained on the production from these machines, it is also essential that there shall be concentricity between the various elements and that the machining shall produce a specified high standard of surface finish. Each machine incorporates an interesting and extremely accurate automatic work cycle.

Three spindle heads are mounted on a common base so that Nos. 1, 3 and 5 valves are machined at the first pass and Nos. 2, 4 and 6 at a second pass. Each head carries three tools. Two tools, that for machining the bore to take the valve guide and that for machining the

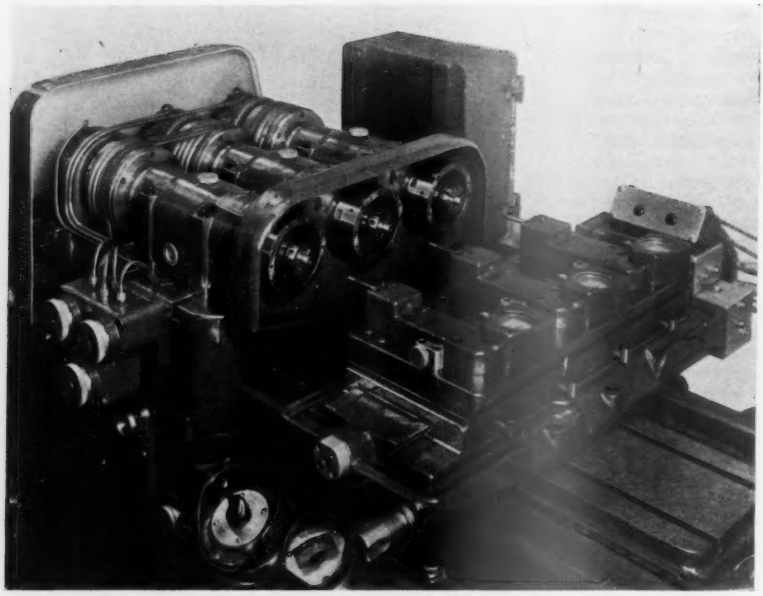


Fig. 6. Set-up for fine boring valve guides on a Precimax machine.

counterbore are mounted in fixed holders. The third tool, for generating the valve seat, is carried in a boring bar which is itself carried in an angular slide in the head. This angular slide is operated by means of a hydraulic operating mechanism at the rear of the spindles.

Apart from its inherent accuracy, the particular interest of this machine lies in the complex automatic cycle. From the loading position there is quick approach of the table to the start boring position, where the table is automatically slowed down to the feed rate and the spindles are started up at the high speed, 1,500 r.p.m. When the boring tool is through the bore for the valve guide, the spindle speed is automatically reduced to 750 r.p.m. and the counterbore tool comes into action. At the end of its forward stroke, the table runs against a positive stop and is then retracted 0.010in. This retraction is included in the cycle to prevent the counterbore tool rubbing against the recess while the third tool is generating the valve seat. This is important for both finish and tool life.

At the end of the seat generating stroke, the angular slide is reversed and at the same time the table, with the component is also reversed. When the reverse stroke of the table has brought the component clear of the counterboring tool, the spindle is automatically increased to 1,500 r.p.m. and a finishing spring cut is taken on the bore to take the valve guide.

As soon as the boring tool is clear of the component, the table with the fixture and component is accelerated to the loading and indexing position. At this point the top slide of the hydraulically-operated cross slide is accurately indexed to the second position to bring Nos. 2, 4 and 6 valve elements into line with the spindles. The cycle is then repeated.

The fixture is carried on the top slide of the hydraulic cross slide and is arranged for through loading. For this reason it is fitted with retracting locating dowel pins. The clamps are hand operated. All the machine controls are centralized on a desk-type push-button station that can be swivelled into the position most convenient for the operator. This control station incorporates two warning lights, one green to indicate that the machine is ready for the initiation of a cycle, and one red that shows while the cycle is in progress. A main selector switch is arranged to select either automatic working of the machine cycle or the setting cycle. In the setting cycle, the automatic features of the machine are inoperative, and each movement can be individually controlled by push buttons. In the automatic cycle, all push buttons are isolated, except for the start and stop button. The floor-to-floor time for finishing all six valve elements is $3\frac{1}{2}$ minutes on each machine.

A third three-spindle Precimax fine borer, see Fig. 6, is used for finish machining the valve guide bores. On this machine, the components are held in double diaphragm chucks which are relieved by hydraulic operating mechanism at the rear end of the spindle for loading and unloading the components. The boring bars are made of solid tungsten carbide, and the tools are brazed into the ends of the bars. Each boring bar is held in a special holder with fine adjustment that can be read on the dial indicators fixed on the machine.

All three tool posts are carried on the top slide of a hydraulically-operated cross slide. This allows the boring bars and tools to be offset 0.0065in from centre when entering the components. At the end of the forward travel of the table, when the bars are at the innermost position, the top slide is hydraulically moved into the boring position for boring to take place on the return stroke.

Red and green warning lamps indicate respectively the cross slide position for the boring and approach strokes. The table movement is interlocked against the cross slide position in such a manner that the table can be moved to the left only when the cross slide is in the rear position and to the right when the cross slide is in the front position. Removing 0.008in of stock from the diameter, the production rate of this machine is 150 components per hour.

Electrical Faults

PHILIPS ELECTRICAL Ltd., Century House, Shaftesbury Avenue, London, W.C.2, have recently introduced an electrical fault finder by means of which breakdowns in the electrical systems of vehicles may be quickly located. The instrument, known as type ER1001, weighs about $8\frac{3}{4}$ lb and is suitable for use on all normal mains between 100 and 250 volts. By means of a series of simple routine tests, voltages may be checked, capacitances measured, insulation tested, and short circuits located. A small neon lamp is provided in the centre of the main panel for use in conjunction with insulation testing. Although designed for use on vehicles, the instrument can also be used for testing many other electrical appliances, as, for example, vacuum cleaners, electric irons, and so on. (1897)

Cemented Carbides

A NEW catalogue of "Cutanit" cemented carbide tools and tips to revised standard shapes and sizes is now available. The revision has been made to reduce very considerably the number of standards and to bring tool and tip shapes into conformity with modern machine shop practice. Users will benefit both from the improvements in design and from the economies in stock cost and maintenance that will result from the reduction in the number of standards. Copies of the catalogue may be obtained from William Jessop & Sons, Ltd., Brightside Works, Sheffield, and J. J. Saville & Co., Ltd., Triumph Steel Works, Sheffield. (1899)

The Dunlop Rubber Co. Ltd., Foleshill, Coventry, have issued a new manual on flexible pipes. These pipes were originally designed for the aircraft industry, but have since found many industrial applications. Full details are given of high, medium and low pressure pipes, and also flame-proof hose.

TRACKED VEHICLES

An Analysis of the Factors Involved in Steering

By W. Steeds, O.B.E., B.Sc., M.I.Mech.E.

(*Continued from page 190*)

Torque and speed relationships

FOR any steering mechanism there are certain relationships between the engine torque (T_e), the sprocket torques (T_o) and (T_i), the steering brake or clutch torques (T_b) or (T_c) and the various gear ratios in the particular mechanism involved, and there are corresponding relationships between the corresponding speeds which will be denoted by the letter ω (omega) with the same subscripts as for the torques. These relationships are given below for several of the mechanisms considered above. The gear ratios involved can also be expressed in terms of the ratio (X) of the sprocket speeds *when there is no slip of the steering brakes or clutches.*

When the brakes or clutches are assumed to be slipping, as has generally been done, in order to make the expressions of more general application, the sprocket speed ratio is denoted by x. In some cases two steering brakes, or a steering brake and clutch, have been assumed to be applied simultaneously although in fact only one would be applied, at least intentionally. This again has been done in order to make the expressions more general. The expressions for the case where the brakes do not slip are easily obtained by putting the brake drum speeds equal to zero or, if it is a clutch that is involved, by putting the speed of the driven element equal to that of the driving element.

The expressions for straight running can be obtained by putting the value of X equal to unity. For all the mechanisms the positive direction for all the external torques applied to the mechanism has been taken as clockwise when viewing the mechanism from the left-hand side, or for longitudinal shafts, from the input end, and speeds have been taken as positive in the same directions. The brake torques (T_b) have been taken as the torques applied to the brake drums, and so it follows that the brake power will always be $-T_b\omega_b$, but since the brake torque will actually be negative when the brake speed is positive and vice versa, the brake power loss will always be positive, as must necessarily be the case, a brake being capable only of absorbing power.

The torques T_o and T_i are taken as the torques applied by the steering mechanism to the sprockets, the final drive ratios being assumed to be unity. The torque T_e is assumed to be the torque applied by the engine to the steering mechanism. The relationships have been given in the form most convenient for the case where the sprocket speeds (or one sprocket speed and the speed ratio) and the sprocket torques are the known quantities and the engine and brake torques and speeds are the unknown quantities, but they can easily be rearranged for the opposite case. When a redundant brake or clutch is assumed to be acting its torque has to be regarded as a known quantity. In all cases the conditions are assumed to be steady, that is the vehicle speed is not changing.

The braked differential system (Fig. 29)

Clearly $\omega_{bo} = \omega_o = x\omega_i$ .. .. (33)

$\omega_{bi} = \omega_i$ .. .. (34)

$2\omega_e = \omega_o + \omega_i = \omega_i(x + 1)$ .. .. (35)

$T_e = T_o + T_i - T_{bo} - T_{bi}$

$= 2(T_o - T_{bo})$ .. .. .. (36)

$T_{bi} = T_i - T_e/2$ .. .. (37)

If $T_{bo} = 0$ then $2T_o = T_e$ .. .. (38)

$T_{bi} = T_i - T_o$ .. .. .. (39)

and power loss in brake $= -T_{bi}\omega_{bi} = \omega_i(T_o - T_i)$ (40)

Taking the numerical example considered previously in which the sprocket speeds were $\omega_o = 10{\cdot}7$ and $\omega_i = 8{\cdot}23$ radians per sec, the track forces $\pm 6{\cdot}65$ tons and the sprocket radius 1·058ft then $T_o = 6{\cdot}65 \times 1{\cdot}058$ tons-ft and T_e is twice this. The engine speed is $\omega_e = (\omega_o + \omega_i)/2 = 9{\cdot}47$ rads/sec. The engine power $T_e\omega_e = 543$ h.p. and the power loss in the brake is 473 h.p. As a check on the working which it is usually useful to make, we have from the principle of the conservation of energy

$$HP_e = HP_o + HP_i + HP_b$$
$$= 306{\cdot}6 - 235{\cdot}8 + 473$$
$$= 543{\cdot}8.$$

This example shows the inherent inefficiency of this system which is not only not regenerative, so that the inner sprocket power has to be dissipated in the steering brake, but which also causes a large proportion of the engine power to be dissipated in the steering brake as well. Of course if the brake is applied so as to lock the inner track, then no power will be dissipated in the brake and the engine will have to develop only the power required by the outer track. Turns under this condition are, however, not practicable except at very low speeds, and there is always the transition stage while the inner sprocket is being brought to rest.

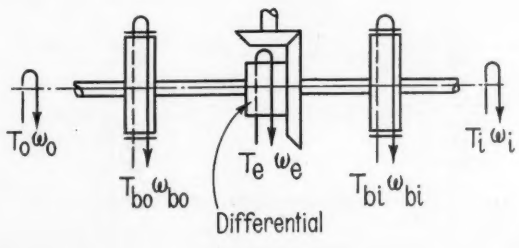


Fig. 29.

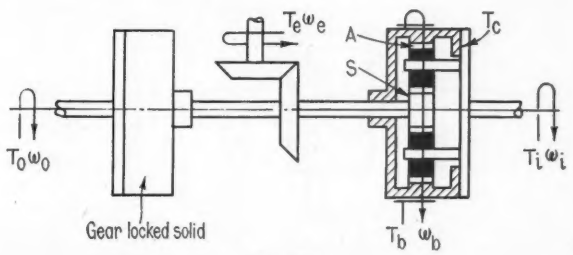


Fig. 30.

TRACKED VEHICLES (Continued)

Geared steering (Fig. 30)

Epicyclic gearing has been assumed as it is the most convenient form to use, but the same expressions will result from any form of gearing giving the same effective gear ratios. The clutch that locks the gear solid for straight running has been assumed to be partly engaged during steering so as to make the expressions more general.

$$\frac{A}{A+S}=m \qquad \frac{S}{A+S}=u \qquad \frac{S}{A}=n$$

ω_e is assumed greater than ω_i and ω_i greater than ω_b.

Then
$$\omega_o=\omega_e \quad .. \quad .. \quad .. \quad .. \quad .. \quad (41)$$
$$\omega_i=u\omega_e+m\omega_b \quad .. \quad .. \quad .. \quad .. \quad (42)$$
$$\omega_b=\frac{\omega_i(1-ux)}{m}=\frac{\omega_i(X-x)}{mX} \quad .. \quad .. \quad (43)$$
$$T_e=T_o+uT_i+uT_c \quad .. \quad .. \quad .. \quad (44)$$
$$T_b=mT_i-uT_c \quad .. \quad .. \quad .. \quad .. \quad (45)$$

If $T_c=0$ then engine power $=\omega_i x(T_o+uT_i)$
$$=\omega_i x(T_o+T_i/X) \quad .. \quad .. \quad .. \quad .. \quad (46)$$
and brake power loss $=-T_b\omega_b=-T_i\omega_i(1-ux)$
$$=-T_i\omega_i(1-x/X) \quad .. \quad .. \quad .. \quad .. \quad (47)$$

Controlled differential steering (Fig. 31)

A spur type differential is shown but, of course, a bevel type will give exactly the same results. $\frac{A.Q.}{B.R}=k.$

Then
$$2\omega_e=\omega_o+\omega_i=\omega_i(x+1) \quad .. \quad .. \quad (48)$$
$$2\omega_{bo}=\omega_i[1-k+x(1+k)] \quad .. \quad .. \quad (49)$$
$$2\omega_{bi}=\omega_i[1+k+x(1-k)] \quad .. \quad .. \quad (50)$$
$$\frac{\omega_o}{\omega_i}=x=\frac{\omega_e(k+1)-\omega_{bi}}{\omega_e(k-1)+\omega_{bi}} \quad .. \quad .. \quad (51)$$
$$X=\frac{k+1}{k-1} \quad .. \quad .. \quad .. \quad .. \quad (52)$$
$$T_e=T_o(1+1/k)+T_i(1-1/k)-2T_{bo} \quad (53)$$
$$T_{bi}=(T_i-T_o)/k+T_{bo}$$

Controlled epicyclic steering (Fig. 32)

$$\frac{A}{A+S}=m \quad \frac{S}{A+S}=u \quad \frac{S}{A}=n \quad \frac{C}{D}=k \quad \frac{E}{F}=h \quad \frac{E}{G}=g$$

Then
$$2\omega_e=\omega_i(x+1)/mg \quad .. \quad .. \quad .. \quad (55)$$
$$2\omega_s=\omega_i(x-1)/uk \quad .. \quad .. \quad .. \quad (56)$$
$$\frac{\omega_o}{\omega_i}=x=\frac{g\omega_e+nk\omega_s}{g\omega_e-nk\omega_s} \quad .. \quad .. \quad .. \quad (57)$$

[If there is no clutch slip then $\omega_s=h\omega_e$ and $X=\frac{g+nkh}{g-nkh}$ hence $\frac{nkh}{g}=\frac{X-1}{X+1}$] .. .. (58)
$$T_s=T_c=(T_o-T_i)uk \quad .. \quad .. \quad .. \quad (59)$$
$$T_d=(T_o+T_i)m \quad .. \quad .. \quad .. \quad .. \quad (60)$$
$$T_e=(T_o+T_i)mg+(T_o-T_i)ukh \quad .. \quad (61)$$

Assuming ω_s to be less than $h\omega_e$, then the power loss in the clutch is $(\omega_i/2)(T_o-T_i)\left[\frac{nkh}{g}(x+1)-(x-1)\right]$
$$=\omega_i(T_o-T_i)\left[\frac{X-x}{X+1}\right] \quad .. \quad .. \quad .. \quad (62)$$

Total engine power $=T_e\omega_e$
$$=(\omega_i/2)(x+1)\left[T_o+T_i+(T_o-T_i)\frac{nkh}{g}\right]$$
$$=\omega_i(x+1)(T_o+T_i)\left(\frac{X}{X+1}\right) \quad .. \quad .. \quad (63)$$

If T_s is zero then $T_o=T_i$, that is, the steering drive bevels act as a balance gear to equalise the sprocket torques just as if a differential were fitted.

Merritt-Brown steering (Fig. 33)

$$\frac{A}{A+S}=m \quad \frac{S}{A+S}=u \quad \frac{S}{A}=n \quad \frac{C}{D}=k \quad \frac{E}{F}=h$$

Then
$$2\omega_e=\omega_i(x+1)/m(nk-h) \quad .. \quad .. \quad (64)$$
$$2\omega_{bo}=(\omega_i/u)[2nkx-h(x-1)]/(nk-h) \quad .. \quad (65)$$
$$2\omega_{bi}=(\omega_i/u)]2nk+h(x-1)]/(nk-h) \quad .. \quad (66)$$
$$\omega_o=u\omega_{bo}-mh\omega_e \quad .. \quad .. \quad .. \quad (67)$$
$$\omega_i=2uk\omega_e-mh\omega_e-u\omega_{bo} \quad .. \quad .. \quad (68)$$
$$\frac{\omega_o}{\omega_i}=x=\frac{u\omega_{bo}-mh\omega_e}{2uk\omega_e-mh\omega_e-u\omega_{bo}} \quad .. \quad (69)$$
$$X=h/(h-2nk) \quad .. \quad .. \quad .. \quad .. \quad (70)$$
$$T_{bo}=(T_o-T_i)u+T_{bi} \quad .. \quad .. \quad .. \quad (71)$$
$$T_e=-(T_o+T_i)mh-2kT_{bi}+2ukT_i \quad (72)$$

The relationships for straight running can be obtained by putting $x=1$ in the speed relationships and $T_{bo}=T_{bi}=0$ in the torque relationships. Usually an interlock is provided to prevent both steering brakes from being applied simultaneously so that ordinarily T_{bi} will be zero. Assuming this to be so, then

Power loss in brake $=-T_{bo}\omega_{bo}$
$(T_{bi}=0)$
$$=\omega_i(T_i-T_o)\left[\frac{2nkx-h(x-1)}{2(nk-h)}\right] \quad (73)$$
$$=\omega_i(T_o-T_i)\left(\frac{X-x}{X+1}\right) \quad (74)$$

Engine power $=T_e\omega_e$
$(T_{bi}=0)$
$$=\frac{\omega_i[T_i(2nk-h)-T_oh](x+1)}{2(nk-h)} \quad (75)$$
$$=\omega_i(x+1)\left[\frac{T_oX+T_i}{X+1}\right] \quad (76)$$

Merritt-Brown steering—neutral turn and reverse

In a neutral turn there is no drive to the annulus shaft, which consequently acts as a balance shaft, so that $T_o=-T_i$, assuming $T_{bi}=0$. Then
$$\omega_{bo}=\omega_i x/u \quad .. \quad .. \quad .. \quad .. \quad .. \quad (77)$$
$$\omega_e=\omega_i(x+1)/2uk \quad .. \quad .. \quad .. \quad .. \quad (78)$$
$$T_e=2T_iuk \quad .. \quad .. \quad .. \quad .. \quad .. \quad (79)$$
$$T_{bo}=2T_ou \quad .. \quad .. \quad .. \quad .. \quad .. \quad (80)$$

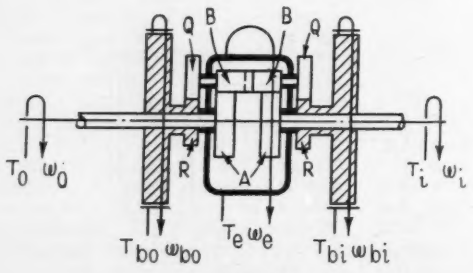


Fig. 31.

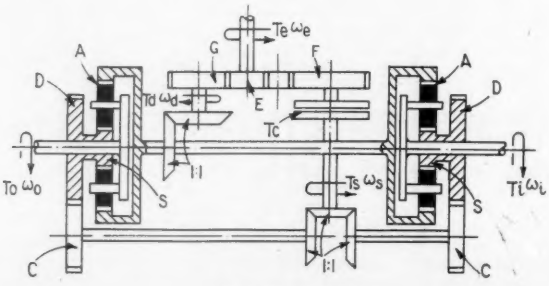


Fig. 32.

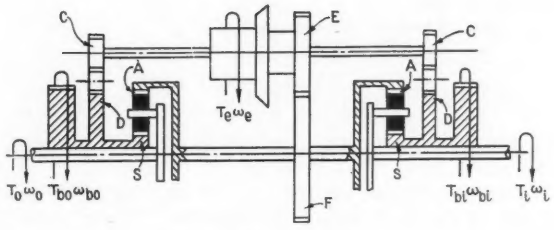


Fig. 33.

To get reverse the annulus shaft is held at rest by a torque T_c applied by a dog-clutch that clutches the shaft to the gearbox casing, then

$$T_c = (T_o + T_i)m \quad .. \quad .. \quad .. \quad .. \quad (81)$$
$$T_e = 2T_i uk \quad .. \quad .. \quad .. \quad .. \quad (82)$$
$$T_{bo} = (T_o - T_i)u \quad .. \quad .. \quad .. \quad .. \quad (83)$$

The "Panther" system (Fig. 34)

In the diagram of this system the reduction given by the gears E and G represents the reduction in the gearbox, and the reduction E to F represents the reduction afforded by the drive to the sun gear when steering.

$$\frac{A}{A+S} = m \quad \frac{S}{A+S} = u \quad \frac{S}{A} = n \quad \frac{E}{F} = h \quad \frac{E}{G} = g \quad \frac{C}{D} = k$$

Assuming $T_{co} = 0$ and $\omega_{bo} = 0$, then

$$\omega_e = \omega_o/mg \quad .. \quad .. \quad .. \quad .. \quad .. \quad (84)$$
$$\frac{\omega_o}{\omega_i} = x = \frac{mg\omega_e}{mg\omega_e + u\omega_{bi}} \quad .. \quad .. \quad .. \quad (85)$$
$$X = \frac{g}{g - nhk} \quad .. \quad .. \quad .. \quad .. \quad (86)$$
$$\omega_{bi} = (\omega_i/u)(1 - x) \quad .. \quad .. \quad .. \quad .. \quad (87)$$
$$T_{bo} = uT_o \quad .. \quad .. \quad .. \quad .. \quad .. \quad (88)$$
$$T_{bi} = uT_i + T_{ci} \quad .. \quad .. \quad .. \quad .. \quad (89)$$
$$T_e = khT_{bi} + T_i(mg - ukh) + mg\,T_o \quad .. \quad (90)$$
$$\text{Engine power} = x\omega_i\left[\frac{kh}{mg}.T_{bi} + \left(1 - \frac{nkh}{g}\right)T_i + T_o\right] \quad (91)$$
$$\text{Brake power loss} = T_{ci}(\omega_{bi} + hk\omega_e) \quad .. \quad .. \quad (92)$$
$$= \omega_i(T_{bi}/u - T_i)\left(\frac{X - x}{X}\right) \quad .. \quad (93)$$
$$\text{If } T_{bi} = 0 \text{ then engine power} = x\omega_i[T_o + T_i/x] \quad .. \quad (94)$$
$$\text{and brake power} = \omega_i T_i\left[\frac{x - X}{X}\right] \quad .. \quad (95)$$

Gates L.G.1 system (Fig 35)

The expressions are derived for the case when the steering, or differential, clutch is slipping and the main clutch on the inside end of the input shaft is not fully disengaged. The other main clutch, however, is assumed to be fully engaged.

$$\frac{E}{F} = h \quad \frac{C}{D} = k$$

$$\text{Then } \omega_e = \omega_o/k \quad .. \quad .. \quad .. \quad .. \quad .. \quad (96)$$
$$2\omega_r = \omega_o + \omega_i = \omega_i(x + 1) \quad .. \quad .. \quad .. \quad (97)$$
$$\text{Slip speed at main clutch} = (\omega_i/k)(x - 1) \quad .. \quad (98)$$
$$\text{Slip speed at steering clutch} = \omega_i\left(\frac{x+1}{2} - \frac{xh}{k}\right) \quad (99)$$
$$\text{hence } \frac{h}{k} = \frac{1 + X}{2X}$$
$$\text{and slip speed at steering clutch} = \frac{X - x}{2X} \quad .. \quad (100)$$
$$T_r = -2(T_i - T_c/k) \quad .. \quad .. \quad .. \quad (101)$$
$$T_e = -T_i(k - 2h) + T_o k + 2T_c(1 - h/k) \quad (102)$$

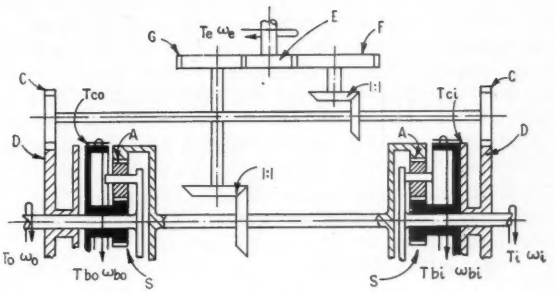


Fig. 34.

$$\text{Engine power} = T_e\omega_e$$
$$= x\omega_i\left[T_o + T_i/X + (T_c/k)\left(\frac{X-1}{X}\right)\right] \quad .. \quad (103)$$
$$\text{Power loss at steering clutch} = \omega_i(T_i - T_c/k)$$
$$\left(\frac{x - X}{X}\right) \quad .. \quad .. \quad .. \quad .. \quad .. \quad (104)$$
$$\text{Power loss at steering clutch} = \omega_i(T_i - T_c/k)\left(\frac{x - X}{X}\right) \quad (105)$$
$$\text{Total power loss}$$
$$= x\omega_i\left[T_i/X - T_i/x + \left(T_c/k\right)\left(\frac{X-1}{X}\right)\right] \quad (106)$$
$$\text{If } T_c = 0, \text{ then engine power} = x\omega_i[T_o + T_i/X] \quad (107)$$
$$\text{and power loss in steering clutch}$$
$$= x\omega_i[T_i/X - T_i/x] \quad .. \quad .. \quad (108)$$
$$= \omega_i T_i\left[\frac{x - X}{X}\right]$$

Comparison of losses when steering brakes are slipped to give larger turning radii

In the steering systems considered above a definite sprocket speed ratio is obtained when the steering brakes, or clutches, do not slip, and this gives a certain steering radius; if the brakes or clutches are allowed to slip then the sprocket speed ratio will have a lower value and the radius of turn will be greater. The slipping of the brakes or clutches involves power losses, and in this section the losses for several steering systems all working under identical conditions will be compared.

Consider two vehicles that are exactly similar in all respects except that one is fitted with geared steering and the other with Merritt-Brown steering, and let both mechanisms be designed so that they give the same sprocket speed ratio when the steering brakes do not slip. Let the brakes now be allowed to slip so that the sprocket speed ratio is reduced to some lower value and let the mean speed of the vehicles be the same.

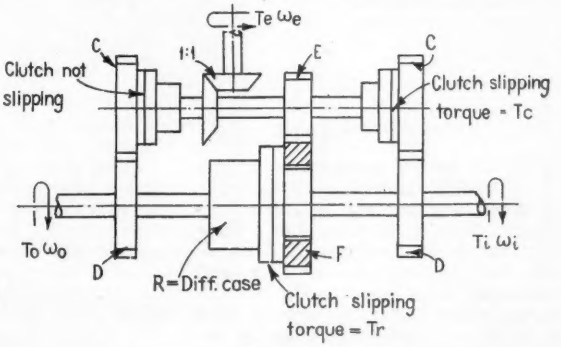


Fig. 35.

TRACKED VEHICLES (Continued)

Then it follows that the actual sprocket speeds for the two vehicles will be the same for both vehicles, but since the sprocket speed ratio is assumed to be the same for both vehicles this condition is satisfied if the inner sprocket speed is the same for both vehicles. Let us now compare the losses. Let $X = \omega_o/\omega_i$ be the sprocket speed ratio when no slip occurs, and $x = \Omega_o/\Omega_i$ be the ratio when the brakes are slipping.

Then for the geared system we have $X = 1/u$
and for the Merritt-Brown system $X = h/(h - 2nk)$
When the brakes are slipping we have

$$\text{Power loss in the geared system} = -T_i\Omega_i(1 - ux) = -T_i\Omega_i(X - x)/X$$

and power loss in the Merritt-Brown system

$$= (T_o - T_i)\Omega_i\left[\frac{2nkx - h(x-1)}{2(nk-h)}\right]$$

$$= \Omega_i(T_o - T_i)\left(\frac{X-x}{1+X}\right)$$

Hence the ratio of the losses is

$$\frac{\text{Loss in the geared system}}{\text{Loss in Merritt-Brown system}} = \left[\frac{-T_i}{T_o - T_i}\right]\left[\frac{1+X}{X}\right] \quad .. \quad .. \quad (109)$$

If the track forces, and hence the sprocket torques, are assumed to be equal and opposite, then the ratio of the losses becomes $(1 + X)/2X$, and for values of X greater than unity, which will always be the condition for turns where the radius is finite, this expression is less than unity, so that the loss in the geared system is less than that in the Merritt-Brown system. This result was first given in the author's contribution to the discussion of Dr. Merritt's paper entitled " The Evolution of a Tank Transmission " when it was presented to the Institution of Mechanical Engineers. It was disputed in a written reply by Lt.-Col. Ewen McEwen, made apparently on behalf of Dr. Merritt on the grounds that it assumed different mean speeds for the two vehicles and so was not a valid comparison. But, as pointed out above, the assumptions are that the inner sprocket speed and the sprocket speed ratio are the same for both vehicles, and this will result in equal mean vehicle speeds and equal radii of turn so that the comparison is, in fact, quite valid. It is true that the engine speed of the vehicle fitted with the geared steering would have to be increased during the turn, but that is of no significance. As a check on this result let us derive an expression for the ratio of the engine powers required in the two vehicles. For the geared system we have :

$$\text{Engine power} = \Omega_i x(T_o + uT_i) = \Omega_i x(T_o + T_i/X)$$

and for the Merritt-Brown system

$$\text{Engine power} = \frac{\Omega_i[T_i(2nk - h) - hT_o](x+1)}{2(nk-h)}$$

so that we get

$$\frac{\text{Engine power, geared system}}{\text{Engine power M-B system}} = \frac{2\Omega_i x(T_o + T_i/X)(nk - h)}{\Omega_i(x+1)[T_i(2nk-h) - hT_o]} = \left(\frac{x}{x+1}\right)\left(\frac{1+X}{X}\right) \quad .. \quad .. \quad (110)$$

If there is no brake slip then $x = X$ and this ratio equals unity as must obviously be the case. When there is brake slip then x is less than X and the engine power in the geared system will be less than that in the Merritt-Brown system. This again must be so if the brake losses in the Merritt-system are greater than those in the geared system because the extra losses must be made good by the engine.

Numerical example

Taking the sprocket speeds to be $\Omega_o = 10{\cdot}7$, $\Omega_i = 8{\cdot}23$ radians/sec, the track forces to be $F_o = -F_i = 6{\cdot}65$ tons and the sprocket radii to be 1·058ft for both vehicles, and assuming that when there is no brake slip both steering mechanisms give $X = 1{\cdot}4$, then when the brakes are slipped to make the sprocket speed ratio $x = 1{\cdot}3$ the following results are obtained.

Power loss in geared system = 16·8 h.p.
Power loss in M-B system = 19·65 h.p.
Ratio of losses = 0·857
Engine power in geared system = 87·6 h.p.
Engine power in M-B system — 90·4 h.p.
Ratio of engine powers = 0·969.

The engine powers may be calculated direct from the expressions developed in the preceding sections or may be computed from the track powers and the brake loss powers. Thus, by the latter method we have :

Engine h.p. = Outer sprocket h.p. + inner sprocket h.p. + Loss h.p.

and for the geared system this gives :

Engine h.p. = 306·6 − 235·8 + 16.8 = 87·6

while for the Merritt-Brown system we get :

Engine h.p. = 306·6 − 235·8 + 19.65 = 90·4.

Assuming that for both vehicles $a_o = -a_i$, as is implied by the assumed values for the track forces, then the mean speed for both vehicles is 10ft per sec. If a_o does not equal a_i, then the mean speed will have some other value (being given by equation 8) but will be the same for both vehicles.

Analysis on the above lines will show that in respect of the losses when the steering brakes are slipped to give larger steering radii the controlled differential and the controlled epicyclic systems are identical with the Merritt-Brown system and that the Panther system and the Gates L.G.1 system are identical with the geared system.

Nimonic Alloys

THREE further publications dealing with the important Nimonic series of alloys have recently been issued by Henry Wiggin & Co., Ltd. They give details of methods for annealing and pickling, hot-working and machining these alloys. The booklet on machining has been compiled from a survey of current practice in the machining of these materials by several prominent manufacturers of aircraft engines.

These publications contain valuable data on the methods to be employed for optimum results. They should be of considerable interest and use to any organization concerned with the fabrication of these materials. They may be obtained, free of charge on application to Henry Wiggin & Co., Ltd., Wiggin Street, Birmingham, 16. (1901)

Rover Diesel Engine

REFERENCES have appeared recently in certain technical journals which imply that Rolls-Royce, Ltd., are responsible for the development of a V-8 engine, based upon the Merlin and which is now being produced by the Rover Company. Rolls-Royce, Ltd., would like to make it clear that they are not responsible for the development of the engine in question.

NEW PLANT AND TOOLS

Some Recent Developments in Production Equipment

A NEW horizontal tool room boring machine the "Optimetric" has been developed by H. W. Kearns & Co., Ltd., Broadheath, nr. Manchester. It is illustrated in Fig. 1 and will be shown for the first time at the Canadian International Trade Fair in Toronto. Primarily, the machine is intended for use in tool rooms and experimental departments for performing surfacing, boring, milling and drilling operations to a high degree of accuracy. It is designed to bridge the gap between the normal high grade machine and the very expensive jig borer, and at the same time be capable of carrying out the operations for which it is designed at one setting of the component where possible.

The combined resources of H. W. Kearns & Co., Ltd., Hilger & Watts, Ltd., and the Production Engineering Research Association of Great Britain were employed in the design, development and testing of this machine. All the mechanical and electrical problems involved were dealt with in the Company's design department, but it was felt that for setting the work in relation to the spindle centre an optical system should be used. The optical system has been developed by Hilger & Watts, Ltd. It is accurate, easy to read, not subject to wear, and will withstand the rigours of workshop use.

Finally, in addition to a thorough test in their own tool room, H. W. Kearns & Co., Ltd., decided to have one tested independently by PERA. Among other items, PERA reported that "ten repeat setting could be made in 60 seconds with an error of repeatability not exceeding plus or minus 0.0002in and that during the tests neither fatigue nor eye strain were suffered."

The optical measuring system is fitted to both the vertical and transverse motions of the machine. It consists of a glass scale mounted in adjustable brackets. This scale has 0.01in graduations engraved on the underside. These graduations are projected on to a large ground glass screen on which are widely-spaced marks of the vernier scale for sub-division to 0.001in. If metric scales are used there is direct reading to 0.1mm.

This system has the following advantages:—

(1) Setting can be made to 0.001in or 0.01mm to an accuracy of 0.00025in or 0.005mm.

(2) The brilliantly illuminated screens, measuring 1½in × 4in, completely eliminate microscope eye-strain.

(3) As the rule is projected on to the side of the ground glass screen on which the vernier is marked, there cannot be any parallax errors in reading.

(4) The built-in scales are of plate glass and are not subject to wear.

(5) Both mountings are adjustable and the rules can be set to any required datum line within a range of 1in.

(6) As the measuring system is built into the machine, measurements are made without the use of end rods, gauges or dial indicators. This reduces the possibility of error and also saves time.

To ensure that full advantage can be taken of the accuracy to which the spindle slide can be set, a special locking system has been devised. It comprises a steel strip on either side of the upright and two locking clamps on the spindle slide. This allows the spindle to be set to position and then locked without in any way disturbing the accuracy of the setting. Another important feature is the built-in facing chuck. This has a facing slide that can be adjusted while the machine is running, thus allowing surfacing and boring operations to be carried out with a simple type of boring bar. A special range of equipment has been developed for use with this machine. The principal dimensions are:—

Size of main table ..	16in × 22in.
Revolving table	18in × 18in.
Longitudinal traverse	18in.
Transverse traverse	12in.
Vertical traverse ..	12in.

Maximum from spindle nose to boring stay, 3ft 3in.

Maximum diameter machine will face, 8in.

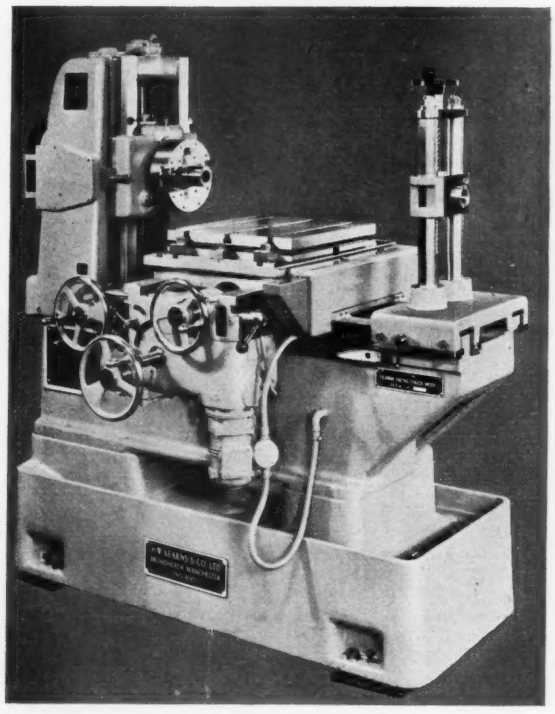


Fig. 1. Kearns "Optimetric" tool room boring machine.

Screw cutting lathe

A new Cardiff 7in screw cutting lathe built by B. Elliot & Co., Victoria Works, Willesden, London, N.W.10 is shown in Fig. 2. It is available in two sizes, 40in and 60in between centres The bed is a once-piece casting, cross-webbed for rigidity. It has a gap that can be covered with a bridge-piece when not in use. The gap is 6⅞in wide, and work 19¾in dia. × 5¾in can be swung.

A heat-treated alloy steel forging is used for the spindle. It is carried in Timken taper roller bearings and is bored 1¾in diameter. The spindle nose accepts a No. 3 Morse taper centre. An attachment for collets up to 1¼in diameter is available as an extra.

A nine-speed gear box in conjunction with a sliding back gear provides 18 spindle speeds over the range 27-775 r.p.m. All the gears are made of alloy steel and the gear shafts are carried in ball races. The whole gear unit runs in an oil bath. Spindle speed changes are made by means of co-axial levers at the front of the headstock cabinet. Initially, the drive to the gear box is by twin vee belts from a 2 h.p. motor, and from the gear box to the main spindle by flat belt. An eccentric pulley is fitted to allow correct belt tension to be maintained on the main drive. The motor is controlled through a third shaft on the bed. The actuating lever is adjacent to the apron.

A Norton gear box facilitates screw cutting. It gives a range of Whitworth threads from 2¾ to 80 t.p.i. and from 0.1875 to 5mm metric pitch. Change over from one thread-system to the other is effected by rotation of a dial on the gear box. An orthodox type screw cutting indicator is flush-fitted on the apron. For the elimination of end play, adjustable nuts and

thrust collars are fitted on the lead screw. Operation of the saddle is by separate rack and pinion through reduction gears.

Both the cross-slide and the top-slide work in dovetail guides fitted with adjustable gib strips to give compensation for wear. Each slide has an easily read graduated dial. The top-slide swivel is graduated 60 deg-zero-60 deg and carries a fully indexing square toolpost. Felt wiper pads are fitted to all slides. The tailstock is aligned by the vee-flat guideways on the bed. It may be set over for slightly tapered work. The tailstock sleeve is graduated in inches, and is bored to take a No. 3 Morse taper centre which is self-ejecting. An electric impeller-type coolant pump with detachable tank can be supplied as extra equipment.

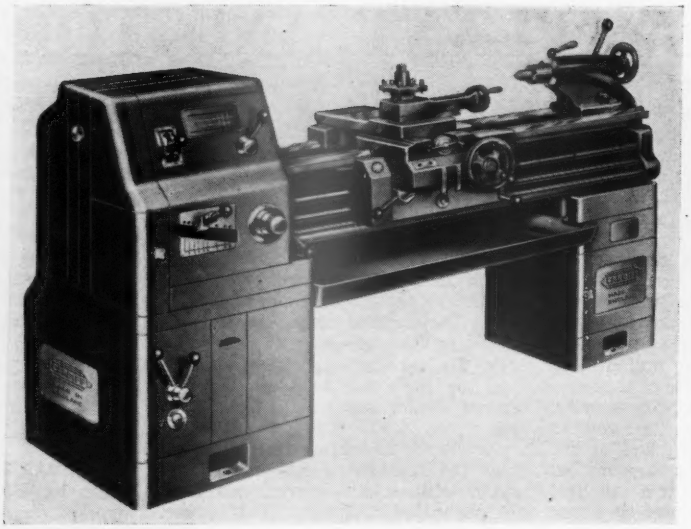


Fig. 2. Cardiff 7in screw cutting lathe.

Coil-fed screw automatic

Hitherto the Datim $\frac{5}{16}$in coil-fed screw automatic machine, which is illustrated in Fig. 3, has been used only in the factory of Davis & Timmins, Ltd., who had developed it specially for their own production purposes. Now, however, arrangements have been made to build these machines for sale in certain overseas markets and The Adam Machine Tool Co., Ltd., 35, Chequer St., St. Albans, Herts, have been appointed sole selling agents.

This machine incorporates a number of unusual design features, all tending to give very high rates of output on precision work. As its name implies, the machine uses coil stock instead of bar stock. This leads to considerable savings in both time and material. For example, a single coil will suffice for the output for which from 50 to 60 bars would be required. The saving in time may therefore, be substantial. In addition, the wastage on one coil will be in the order of a 36in end piece, whereas there will be a minimum wastage of 3in end piece per bar with a total wastage of more than 12ft on 50 bars.

The coiled wire is automatically straightened by a powerful wire straightening attachment before it passes through the machine. A patent clutch operating mechanism is incorporated to ensure that the exact length of wire is fed forward. When the coil is exhausted, or if the wire breaks in the straightener, the machine is stopped by an auto cut-out. Before it enters the straightener, the wire is cleaned and lubricated by a self-contained cleaning box.

The machine has a sliding head which moves longitudinally on the bed. It carries the non-rotating work spindle and collet operating mechanism. The work spindle supprts the collet and the feed finger tube, which is anchored to the bed. Feed to the head is controlled by a bell cam, and the collet is operated by adjustable cam dogs. The stock is rigidly held by the collet chuck incorporated in the sliding head work spindle, and by a wire grip mechanism attached to the bed. While the parting off operation is being completed, the sliding head is returned to the feeding position. This effects a considerable saving in idle time.

A revolving tool spindle is employed. It carries two cutting tools. A single bell cam controls the operating movement of the cutting tools through hardened and ground spiral splines. The tool spindle also supports the wire guide bush holder in a position immediately behind the cutting tools. This gives maximum support to the wire, and ensures a high degree of finish, accuracy and concentricity.

The tool holder plate is carried by the revolving tool spindle. It consists of two simple tool holder blocks, precision made and hardened and ground. To relieve the operating mechanism of undue strain from centrifugal action when revolving at high speed, a patented compensating mechanism is incorporated in the tool plate. Normal standard design cutting tools are used. They are made to dimensions that allow for accurate setting with the minimum of skill.

Drive to the camshaft is through a gear box and worm gear. Change gears in the gear box are used for controlling the camshaft speed. An adjustable safety clutch minimizes the risk of breakage in case of overload. Provision is made for both manual and power drive to the camshaft. All the cams are easily fitted and can be changed without disturbing the camshaft. In addition to the wire straightening attachment, there are specially designed screwing, slotting and drilling attachments. The principal dimensions are:—

Maximum length of part produced, 1in.

Maximum screwing dia., $\frac{3}{16}$in.

Maximum length of thread with standard die holder, $\frac{3}{8}$in.

Range of camshaft cycle times, approx., 1 to 5 seconds.

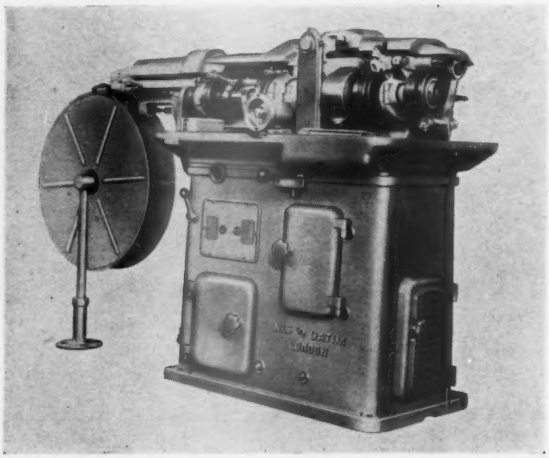


Fig. 3. Datim $\frac{5}{16}$in coil-fed screw automatic.

Gear hobbing

A recent addition to the range of gear hobbing machines designed by Dowding & Doll, Ltd., Greycoat St., London, S.W.1, is illustrated in Fig. 4. It is designed for a wide range of work, including spur and helical gears, serrations, ratchets and chain sprockets. If desired, it can be fitted with an auxiliary tangential hobbing head for producing worms and worm wheels. It is claimed that this attachment allows the hobbing of larger and wider high-angle worm wheels than are normally accommodated on comparable machines.

Close-grained cast iron is used for the main casting which is adequately ribbed

and the whole machine is designed to have great rigidity and freedom from vibration. A single-start master worm wheel of phosphor bronze is used to ensure accuracy of pitch in the index drive to the work spindle. The mating worm is of nitrided steel, hardened and ground. This worm is mounted in journals independent of its splined shaft, and has both axial and radial adjustment. The indexing unit is mounted in a totally enclosed oil-bath housing.

The drive to the main spindle is by single vee belt from a 2 h.p. motor. Drive to the hob spindle is taken from the main spindle through pick-off gears, splined vertical main-shaft spiral bevels and helical gears. The spindle is of robust design and is hardened and ground. It runs in phosphor-bronze bearings and provision is made for axial adjustment. The pick-off gears give 10 speeds in the range 61-246 r.p.m. To give smooth and shockless drive to the hob, a balanced flywheel is mounted at the lower end of the fast running vertical main shaft. The hobbing head, which can be swivelled through 220 deg to accommodate right or left-hand helical gears, is carried on a vertically traversible saddle. This saddle is mounted on vee guide ways and is actuated by an Acme form lead-screw and nut of guaranteed accuracy.

Any number of teeth between 6 and 400, with the exception of certain prime numbers, can be cut with the standard index gears. A special set-up for a prime number can be supplied. Indexing is effected through change gears, an adjustable quadrant and a splined index worm-shaft. Feed gears control both the hobbing saddle for vertical traverse and the table saddle for radial in-feed. Feeds from 0.005in to 0.120in can be obtained. At the end of the required traverse, the feed may be stopped either automatically or by hand control.

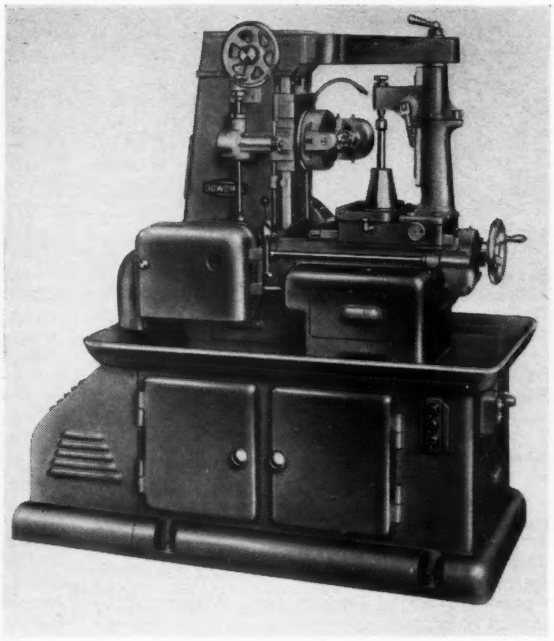


Fig. 4. Dowding and Doll gear-hobbing machine with tangential head.

The machine incorporates a differential unit together with a complementary system of change gears for hobbing helical gears, worms and worm wheels. This differential unit ensures that the spiral angles of mating gears are absolutely the same, irrespective of their face width, and that the gear axes are parallel. In addition, the differential mechanism allows the feed to be changed without influencing the lead or involving recalculation. A further advantage is that a finishing cut may be taken without losing the pitch position of the hob.

Automatic contour turning

An important development in machines for automatic contour turning is illustrated in Fig. 5. This machine, a Keller-controlled roll turning lathe has been developed by The Monarch Machine Tool Co., Sidney, Ohio, U.S.A. It is essentially a heavy duty machine designed to give very much higher output rates than are possible on ordinary roll turning lathes. Provision is made for 24 spindle speeds over the range 2½-505 r.p.m. The spindle has a 15in flanged nose and runs on double precision taper roller bearings. All the gears are of the helical type and are either hardened and shaved or hardened and ground. The spindle and all shafts are hardened and have ground threads. A combined force and splash lubrication system supplies the bearings with filtered oil in metered quantities.

A special heavy duty design tailstock is employed. It has bearings equal in size to those fitted in the headstock, and consequently heavy cuts can be taken towards either the headstock or the tailstock, whichever is the more convenient. These bearings carry a 4⅝in diameter rotating quill. The final tightening of the quill is effected by means of a large capstan handwheel. An additional handwheel at the front end of the tailstock is provided as an auxiliary means of moving the quill.

The apron has a built-in reservoir from which lubricant is supplied to all working parts of the apron, the cross-feed screw, the cross slide and the carriage ways. When a D.C. adjustable main motor is used, electric apron control can be supplied to give the operator complete speed control from the apron. It is so arranged that movement of the control lever gives any desired speed forward or reverse and starts or stops the main motor. An individual motor driven rapid traverse

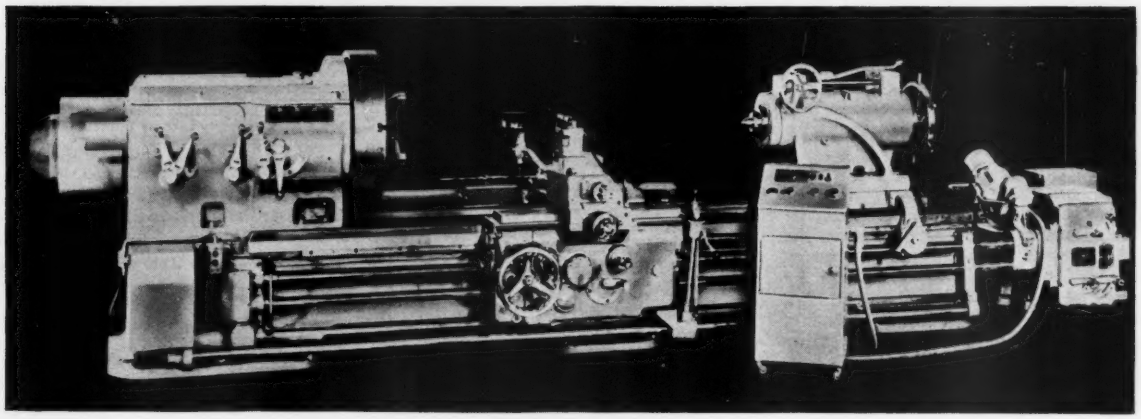


Fig. 5. Keller-controlled Monarch roll-turning lathe.

mechanism, integral with the Keller drive is incorporated in the design. It gives movement in either direction independent of the longitudinal feed.

The Keller control mechanism consists of a magnetic clutch drive and feed change box at the right hand end of the lathe bed, with the tractor mechanism mounted at the rear of the bed. As is normal with Keller-controlled machines, the stylus of the tracer mechanism follows the outline of a thin sheet metal template. One template is used for both roughing and finishing operations. The stylus has three positions determined by very slight pressure changes against the template. One position creates feed towards the centre of the machine, the second gives feed parallel to the machine axis, and the third feed away from the centre. As variations in template contour change the pressure of the stylus against the template, magnetic clutches are energized to actuate the cross and longitudinal screws in the rotary motion. These magnetic clutches are energized or de-energized in 1/125th of a second, so that response to change in template contour is practically instantaneous.

Many advantages are claimed for this machine in comparison with conventional roll turning methods. In the first place, the desired form is produced accurately and automatically and all rolls produced from one template will be exactly similar. At the same time there is an amazing reduction in machining times. In one application the time for turning a chilled cast iron roll was reduced from 7 days to 4½ hours. Furthermore, as single point turning is employed, expensive form tools are not necessary. Rockwell Machine Tool Co., Ltd., Rockwell House, 2nd Way, Exhibition Grounds, Wembley, Middlesex, are the agents for these machines.

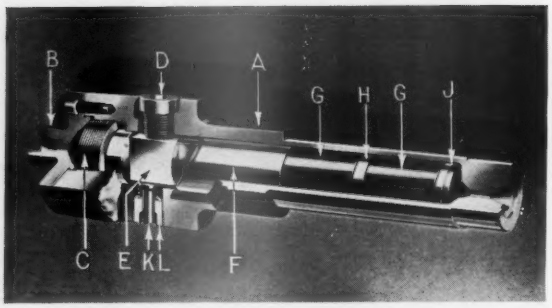


Fig. 6. Herbert-Bristol expanding mandrel.

Expanding mandrels

Alfred Herbert, Ltd., Coventry, have been appointed sole licensees for the manufacture and sale of the rubber expanding mandrels developed and patented by The Bristol Aeroplane Co. A section view of a Herbert-Bristol expanding mandrel is shown in Fig. 6. The design is based on the fact that rubber is incompressible, that is, while it changes its shape under pressure its volume remains constant. Briefly, the design is such that a thrust is exerted on the expanding medium, commercial rubber, and thence transmitted to the expanding portion of the tool.

Fig. 6 shows the construction. Two rubber expansion plugs G and end plug J, a distance piece H and a pressure pad F fit closely in the bore. The screw D actuates the wedge E and transmits pressure to the rubber plugs, the thrust being taken by the screwed plug C. Owing to the constant volume factor of the rubber, the mandrel is expanded by an amount restricted only by the elastic limit of the material from which the mandrel is made. The maximum recommended expansion is 0.002in. Over-expansion is prevented by the screw D, the head of which comes into contact with a shoulder. When the screw D is released, the wedge E is returned by the spring-loaded plunger K which is retained by the screwed plug L, and the mandrel contracts to its original diameter. Access to the components of the mandrel is effected by removing the cap B.

Experience has shown that this equipment has several advantages. It gives extremely close control of production limits on repetition work, since the work can be slipped on to the mandrel up to a collar or stop to ensure that successive work pieces are mounted in exactly the same longitudinal position. As the mandrel is not in any way forced into the work, wear is not caused on either. This means that a Herbert-Bristol mandrel has a much longer useful life than the conventional taper mandrel. Alfred Herbert, Ltd., are prepared to grant sub-licences to organizations wishing to make these tools for their own use.

G.E.C. Spot Welder

A NEW 4 kVA spot welder, type A, has been developed by The General Electric Co., Ltd., Magnet House, Kingsway, London, W.C.2, for the resistance welding of very thin sheets and wires, or for any purpose for which precise control of all the variable factors is of importance. Hitherto many non-ferrous metals and alloys have been considered unsuitable for resistance welding, but this machine with its short, accurately controlled welding times and fully controllable current and electrode pressure makes it possible to weld these metals successfully. High conductivity metals such as copper and silver, which are naturally difficult subjects for resistance welding, can be welded satisfactorily by providing the weld current in the form of a heavy pulse of one cycle duration, and by using electrodes of a relatively high resistance metal such as molybdenum.

Short, accurately controlled welding times are obtained by an electronic control circuit which connects the primary of the welding transformer to the mains through two reversely-connected thyratrons. A resistance-capacitance combination in the grid circuit of the valves controls the welding time accurately between 1 and 6 cycles. The setting of the variable resistance determines the welding times, while the use of a synchronized circuit ensures that the weld always starts at the same part of a cycle.

Electrode holder

It will be appreciated that with a welding time in the order of 1/50th second, the collapse of the metal is extremely rapid and a quick follow-up by the electrode is essential to give proper forging of the weld. For this reason a low-inertia electrode holder has been designed. Welding current is adjustable by means of a variable resistance in the primary circuit of the mains transformer. Electrode pressure is variable between 5 and 50 lb and the foot-operated switch for welding will function only when the pre-set pressure has been applied. The electrodes can be spaced to allow components to be welded to relatively large assemblies without difficulty.

In the case of non-ferrous metals and alloys, welding is successful only when the conditions are exactly right within very close limits. For this reason fully variable controls are essential. This point has been fully borne in mind in the design of this new welder, and records can be kept of all settings with the knowledge that similar settings at a later date will produce similar results. Other features of this machine include a signal lamp to indicate when the weld has been completed and a built-in work lamp with an adjustable reflector.

(1902)

Readers will note a slight reduction in the number of pages in this issue of *The Automobile Engineer*. This reduction has been necessitated by the withdrawal of overtime working by a section of the printing industry and has, in the circumstances, been unavoidable. We hope that it will be only temporary.

PILOT INJECTION

A Description of the Equipment Manufactured by C.A.V. Ltd.

FOR some considerable time research has been in progress on the subject of knock in compression ignition engines, and the influence of the delay period on the combustion of fuel. As is well-known, the delay period is the cause of the uncontrolled burning of the first part of the fuel injected into the combustion chamber, giving rise to an excessively rapid pressure rise and the consequent characteristic noise known as diesel knock. The rapidity of the pressure rise when active combustion begins at the end of the delay period depends largely upon the amount of fuel that has been injected during that time. If the delay period can be shortened or if the rate of fuel

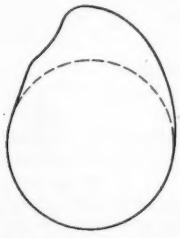


Fig. 1. Pilot injection cam profile.

injection can be reduced during the delay period, the rate of pressure rise will be reduced, with reduction in shock loading and engine noise.

Shortening of the delay period requires, in a given engine, an improvement in the ignition quality of the fuel, that is to say, increase in the cetane number. It is probable, however, that economic conditions will be unfavourable to any improvement in the cetane number of diesel fuels and a second alternative, namely the reduction of the initial rate of fuel delivery, is more likely to provide a practical solution. In general this solution has been known as "pilot" injection.

There are two methods of obtaining the desirable low initial rate of fuel delivery without impairing the efficiency of the combustion cycle consequent upon a reduced rate throughout the injection period. The first method is to inject a small quantity of fuel ahead of the main charge. This pilot charge begins to burn after the delay period, but the quantity is so small that the amount of uncontrolled pressure rise is less than normal. The main charge is timed to inject just as combustion of the pilot charge begins, and since the flame has already been propogated, it is not subjected to any appreciable delay and it burns smoothly under the control of the chosen rate of injection. This rate is high enough to allow combustion to be completed quickly enough for the maximum cycle efficiency to be retained.

The second method, which is now being applied to high-speed passenger transport engines by C.A.V. Ltd., Acton, London, W.3, avoids the mechanical complications of the distinct break in the injection involved in the first method. It provides a continuous injection, the first part at a low rate lasting for the delay period, followed without interruption by the second part at a higher rate. The quantity of fuel injected during the delay period is therefore kept at a minimum, while the remainder is injected at a high rate to complete combustion as rapidly as possible but still under control and retaining the full cycle efficiency.

A two-stage cam, Fig. 1, gives the desired rate of injection characteristic and provides a plunger displacement curve of the form shown in Fig. 2. The pumping plunger is of the C.A.V. port-controlled type with helical spill edge giving termination of injection according to load. Pumping begins when the cam follower approaches the top of the first stage of the profile and continues at low velocity over the nose on to the second stage of the cam. This phase is shown by A on Fig. 2. On reaching the flank of the second stage of the cam the follower is lifted abruptly and pumping continues at a much increased rate until the injection ends, when the spill port is uncovered by the helical edge of the plunger. This phase is shown by B on Fig. 2.

In practice the low initial pumping rate does not result in a correspondingly low rate of fuel injection from the nozzle unless the nozzle is arranged to open at a much lower pressure than that normally employed. The reason for this is evident from the fact that compression of the fuel in the delivery pipe must take place before a pressure high enough to open the nozzle is obtained, and the energy thus stored produces a sudden release of fuel from the nozzle when it eventually opens. If the usual differential valve nozzle is adjusted to open at a fuel pressure low enough for control of injection to be maintained, the pressure at which it closes at the end of injection is even lower and will be less than the gas pressure in the engine cylinder. Consequently the nozzle valve would be held open by the gas pressure until this had fallen during expansion, and hot gases would enter the nozzle, causing fouling.

This problem is overcome by the special type of nozzle shown in Fig. 3. The nozzle needle has two lapped guide portions A and B forming with the nozzle body two separate spaces C and D in which fuel pressure can operate. The upper space C is in direct communication with the fuel feed passage E via the annular groove F in the lower face of the nozzle holder body, and the lower space D communicates with the same feed passage via a small

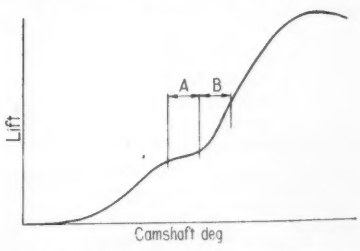


Fig. 2. Pump plunger displacement diagram.

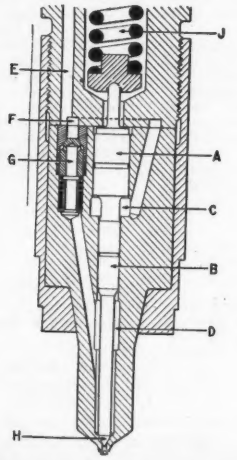


Fig. 3. Section of injector nozzle.

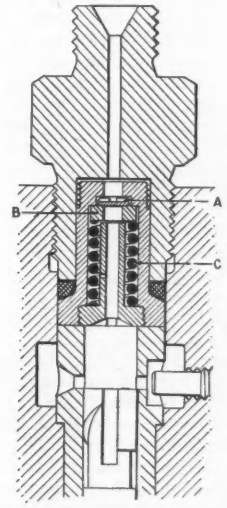


Fig. 4. Pump delivery valve.

PILOT INJECTION (Continued)

non-return valve G. The lower space also communicates via the main nozzle valve seat H to the injection holes in the usual manner. The valve is held down by the spring J. When pressure rises in the feed passage E at the beginning of injection, it operates equally in both spaces C and D since the non-return valve offers negligible resistance to flow of fuel in this direction. Thus the effective area of the needle valve exposed to fuel pressure during the opening phase is equal to the difference between the area of the large guide diameter A and the valve seat H, the small guide diameter B playing no part during opening.

Compression of the spring J is adjusted to give the required opening pressure, usually 70 atm., and is equal to the product of this pressure and the effective area referred to above. When the nozzle valve has lifted, the effective area upon which fuel pressure operates is increased by the area of the valve

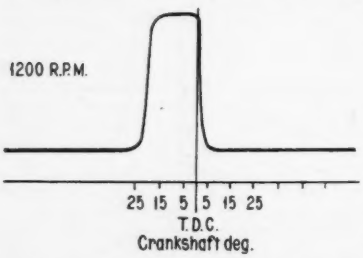


Fig. 6a. Needle valve lift diagram, normal injection.

seat H. It is this increase in effective area which in the normal simple differential valve nozzle causes the closing pressure to be lower than the opening pressure. During the closing phase following the uncovering of the spill port of the pump by the helical edge of the plunger, fuel pressure in the feed pipe and passage E begins to fall and with it the pressure in the upper space C. This fall in pressure in the upper space causes the nozzle valve to begin to close under the influence of the spring force. No corresponding fall in pressure can take place in the lower space D, since flow of fuel back to the feed passage is arrested by closure of the non-return valve G. Fuel from this space thus continues to be expelled from the nozzle holes at a pressure determined by the downward displacement of the needle valve under the influence of the spring. Pressure at the nozzle holes is maintained higher than the cylinder gas pressure until the final closure is made by the seating of the main nozzle valve, thus avoiding any possibility of fouling.

The essential difference between the simple differential valve nozzle and the pilot injection nozzle is that in the case of pilot injection, closing of the valve is not brought about by fall in pressure of fuel in the space immediately in communication with the nozzle holes, but by fall in pressure in the supplementary space temporarily isolated

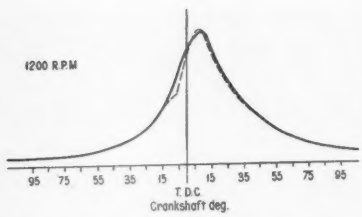


Fig. 5. Typical cylinder pressure diagram.

from the nozzle holes by closure of the non-return valve.

Also associated with the low opening pressure of the nozzle is the design of the pump delivery valve. Although having a retracting piston to avoid undesirable residual pressures in the fuel line, the normal pump delivery valve cannot re-open once it has shut on the termination of injection before the next pumping cycle. Therefore, the pressure wave set up at the nozzle by the closure of its valve is almost completely reflected at the closed delivery valve and travels forward again to the nozzle with possibly sufficient intensity to re-open the nozzle, particularly if the nozzle is set at a low opening pressure.

The valve used with pilot injection equipment shown in Fig. 4, avoids complete reflection of pressure waves. During the forward flow of fuel from the pump to the nozzle the valve offers little resistance since the small plate valve A is free to lift off its seat. When the pump pressure falls at the end of injection, reverse flow causes the valve A to re-seat, but unloading of the pipe line continues by opening of the larger annular valve B downwards against the spring C. This valve will remain open until line pressure has fallen to the pressure determined by the spring loading. Pressure waves arriving at the delivery valve will therefore not be completely reflected even if this return flow valve has temporarily shut and so the intensity of return pressure waves arriving at the nozzle will be considerably reduced and will be unable to re-open the nozzle.

Typical cylinder pressure diagrams are shown in Fig. 5 in which the con-

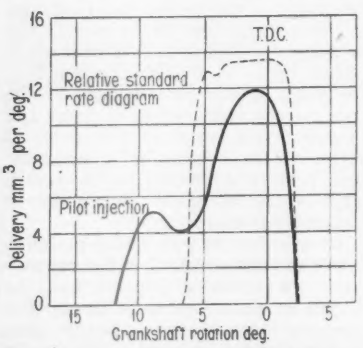


Fig. 7. Curve showing fuel injection rate.

tinuous line represents the compression-combustion curve with pilot injection. The usual high rate of pressure rise following the initiation of combustion has been replaced by a smooth transition from the compression curve.

A diagram from the same engine with normal injection is superimposed and is shown by the dotted line. Typical injector needle valve lift diagrams are shown in Fig. 6, that at *a* showing normal injection, whereas at *b* is shown pilot injection. The opening of the needle corresponding to the first portion of the cam lift is clearly shown. Fig. 7 illustrates fuel injection rates obtained by means of an injection rate analyzer apparatus developed by C.A.V. Direct measurement can be made of the amount of fuel delivered by a nozzle during each degree of camshaft rotation. The full line shows delivery for pilot injection and the relative standard rate diagram is superimposed and shown by the dotted line.

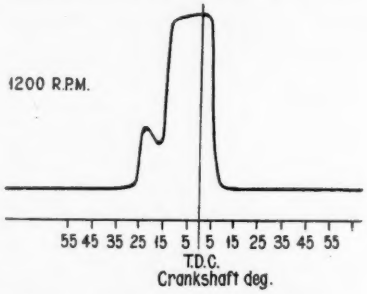


Fig. 6b. Needle valve lift diagram, pilot injection.

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THE FIAT DIESEL

A 50 b.h.p. Power Unit for a New Italian Tractor

TRACTOR manufacture is a post-war development at the Fiat Works, and the organization has recently been largely re-equipped with up-to-date machine tools. The tractor is of the crawler type with a belt pulley horsepower of 45. A five-speed gear box is fitted, giving the following speeds and drawbar pulls:—

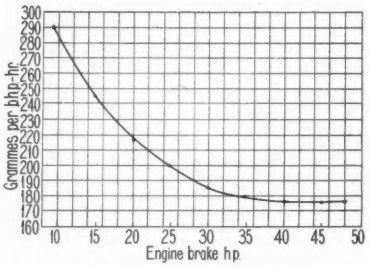
Gear	*Miles per hour*	*Drawbar pull*
1	1.4	10,230 lb
2	2.24	6,270 lb
3	2.8	4,730 lb
4	3.5	3,630 lb
5	5.28	2,310 lb

Each track has five bearing rollers and the shoe widths may be 13¾in, 17¾in or 25½in corresponding to ground pressures of 5½ lb per sq in, 4½ lb per sq in or 2·84 lb per sq in respectively. In ploughing order the total weight is about 10,890 lb.

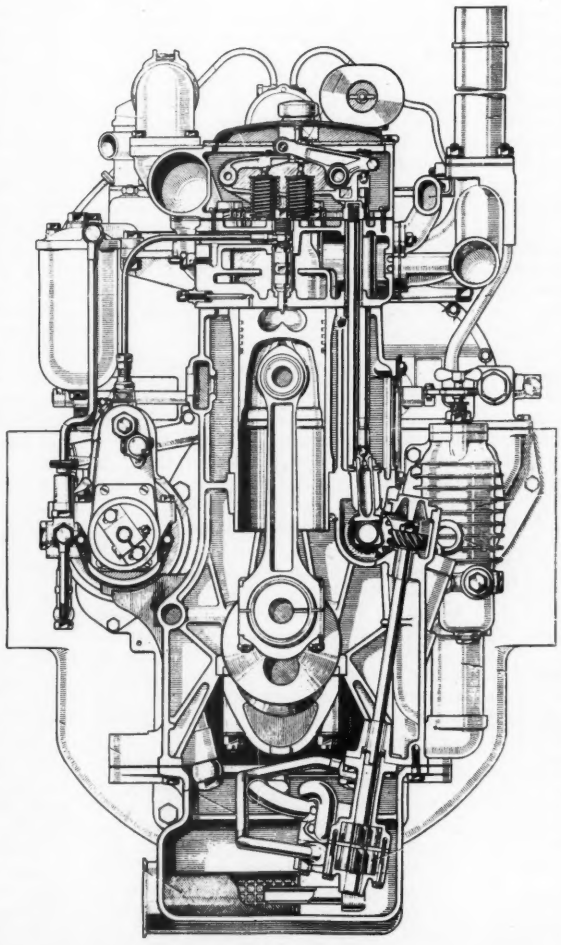
The engine is a four-cylinder four-stroke diesel having a bore and stroke of 100 mm by 140 mm, giving a swept volume of 5,320 c.c. and developing a maximum of 50 b.h.p. at the governed speed of 1,400 r.p.m. There are four valves per cylinder, mounted vertically in the cylinder head and operated by push rods and rockers from hollow mushroom tappets carried in the crankcase. Each pair of valves is operated by two rockers, the primary rocker having a ball-ended adjusting screw at the push rod end with a roller carried on a pin in a fork at the opposite end. Inboard of the roller is a second adjusting screw, the lower end of which makes contact with one valve stem.

Pivoted from the opposite side of the cylinder head is a secondary rocker carrying a ball-ended screw, the tip of which bears on the stem of the other valve, and the rocker is extended beyond the valve to form a bearing face for the roller of the primary rocker. The injector nozzle is mounted centrally between the four valves.

An open chamber combustion system is embodied and the pistons have toroidal cavities. They carry four compression and one scraper ring and the gudgeon pin is located by internal circlips. The H-section connecting rods are of conventional form and the double-ribbed big-end cap is retained by two bolts. Wet cylinder liners are fitted.



Specific fuel consumption.



Fiat 50 b.h.p. diesel engine.

Some interest attaches to the crankshaft design, since the crankpins and journals are bored. The oil ways are therefore tubed across the journal bores but the crankpins, instead of being closed by the customary end caps and through bolt, are provided with light section liners of smaller outside diameter than the bore of the crankpins and having their ends swaged out to the bore diameter so that when in position they provide an annular space into which break the drillings from the main journals, and from which the drillings to the big-ends are taken.

The front of the crankshaft carries a helical pinion which drives the camshaft via an intermediate gear, while a pulley flange mounted on a tapered hub, secured to the crankshaft by the starting handle nut, provides the drive for the fan, water pump, and dynamo by means of duplex V-belts. The gear-type oil pump is driven from the camshaft by spiral gears and an extension shaft.

Mounted low down on the side of the crankcase, the fuel injection pump is also gear driven from the camshaft and the high pressure delivery pipes are led to the injectors through tunnels in the cylinder head. Above the bell housing is a two-cylinder 10 b.h.p. petrol engine. This is the normal means of starting the main engine and it is itself started by pulling a cord wound around its flywheel after the manner of an outboard engine.

The radiator is carried by brackets formed integrally with the bottom tank and flange mounted on faces at the front of the crankshaft. A shroud is provided for the four-bladed fan and the system embodies a by-pass thermostat. Although the impeller spindle is carried in a double row ball bearing at the fan pulley end, the impeller end runs in a plain bearing. An adjustable gland is provided.

In average conditions the consumption of gas oil is stated to be 14·3-16·5 lb per hour.

MATERIAL HANDLING*

A Resumé of General Principles

EVEN now, some managements that are normally progressive and well informed are late, and occasionally reluctant, to recognize that the planning of material handling is primarily the responsibility of the production planning engineer. Since the war, managements are showing increased concern about high inventory figures, without always appreciating that the methods of production virtually determine the amount of material that is necessary to fill the factory pipe-line.

So often the uninformed are apt to relate material handling only to stores and warehousing and the inter-departmental movement between the various producing areas. Although this item is of great importance and major proportions, material handling should cover a vastly greater field. Frequently it starts in the factory of the raw material supplier and is finalized only when the product reaches the customer. An efficient system aims at a streamlined flow of productive material through the plant. It also takes cognisance of the physiological limitations of the operators and should help to create safe working conditions. Further, it is affected by the manner in which the part is processed and the number of times the production operator physically handles the component during the various stages of productive operations.

The evils of an inflated inventory are too well known to need labouring. Yet how often have the savings obtained by buying ahead of schedule to reap the benefits of a favourable market been offset by the additional cost of storage with its attendant high material handling charges. In fact one very large firm contemplated the allocation of a separate warehouse to accommodate material overflow in order to prevent silting up the factory pipe-line with excess stocks. It was proposed to debit the supply department with the cost of storage against each purchased item. The responsible authority predicted that the warehouse would be empty in less than twelve months.

Inventory stocks

It is a fundamental of good business to establish the most rapid stock turn-over in order to get the greatest return of invested capital. However, no matter how well-planned the material handling technique, it will inevitably fail if through lack of adequate material and production control systems, the material pipe-line becomes silted up. In these days of continual striving for higher output at lower unit cost, it is well to remember that this is most likely to be obtained by accelerating the flow of work through the plant. Good planning accomplishes this by better production techniques, keener control systems and a fuller understanding of the needs of material handling. It will certainly not be accomplished by any method that both builds up the operating personnel and increases the "in process" banks to a degree where the material pipe-line becomes clogged.

It must be stressed that the method of production is related to the amount of stock carried. When a decision has to be made on the advisability or otherwise of installing a machine or a piece of productive in order to eliminate a bank of "in process" material, it is purely a question of economics. For example, the productive capacity of a single machine may be such that in order to utilise it to the full, more than one type of component must be produced on it. Immediately this is done, an interruption is caused to the material flow. Therefore, banks of material, in quantity in exact proportion to the period of interruption, must be established. The cost of these banks of material, plus the cost of handling equipment and floor space necessary to contain them and the handling time involved could in some instances outweigh the cost of an additional machine, particularly when amortization is spread over ten years. If banks of material are established, handling and inventory costs are inevitably increased.

Handling methods

Where there is flow production, the conveyor can be a most economical material handling tool. If the production is of the interrupted class, the unit load inevitably comes into its own. There is no doubt that when a conveyor can be loaded by an operator with something he has produced, immediately it is produced, and can conveniently be unloaded by an operator needing the part, the conveyor is operating at near-maximum efficiency.

To-day the comparative methods of material handling by power-pulled conveyor or by fork truck unit load palletization seem to have developed into a controversial subject. This should not be so. Each method has its own inherent advantages, but it is true that the introduction of the unit load technique has considerably reduced the sphere of application for conveyors. Nowadays it is very difficult to justify the installation of a conveyor, if it has to be loaded and unloaded by personnel other than the actual production operators. In other words, if personnel are employed solely for loading and unloading material conveyors, it may be assumed that the conveyor is not particularly efficient.

Pre-planning

Several factors must be considered in determining the quantity of material that will be carried in the factory pipe-line. Before any planning of material handling methods can be accomplished it is necessary to know the volume to be handled and how that volume is established. Recent experience in planning the Vauxhall factory extensions has shown that until the production techniques are established, it is impossible to determine the amount of storage area necessary, or to decide upon the nature and quantity of handling equipment that will be required.

For example, the location of the inspection point for raw materials can have an important influence upon the amount of stock that must be carried. If this inspection is carried out in the supplier's factory, the material upon receipt is available for immediate use and the amount of stock required is less than when incoming goods are inspected after receipt. Inspection at the supplier's factory also reduces the amount of handling necessary, especially where the unit load technique is employed.

Secondly, consider the effect of producing two different components on the first machine, in a line so that there is maximum machine utilization. This inevitably increases the "in process" stock bank, but in what degree depends upon other factors. In laying out a production line on this basis, the production engineer must carry out a complete analysis before any decision is made. If, for example, the machine could produce the alternate components without there being need for re-setting, and it had at least twice the capacity of subsequent machines, an intermediate bank would not be necessary, providing the second machine line was continuous.

If, however, the alternate component is to be used in an assembly remote from the producing machine, then a bank of components proportionate to the trucking frequency is necessary. If a conveyor is used, the bank is proportionate to the speed and spacing of the conveyor. It is clear that consideration of material handling problems by the production engineer, will actually influence the type of machine to be used and the type of jig or fixture to be used with the machine.

The art of analysing a production process or a series of production operations in order to ensure the most economical set-up for material handling has been brought to a very advanced stage

* From a paper presented to the Materials Handling Conference of the Western Section of the Institution of Production Engineers by H. P. Mott, Vauxhall Motors Ltd.

in American industry. For this purpose the American Society of Mechanical Engineers has introduced a standard type of flow process chart, on which are five operational symbols, one each for operation, transportation, inspection, temporary storage and permanent storage. Reduced to essentials, the flow process chart is nothing more nor less than a handy form on which to tabulate "what happens" to a piece of material, component, sub-assembly or other identifiable unit as it passes through a sequence of processes or operations. We may well benefit by the introduction of such charts.

Production tempo

Under certain conditions of manufacture it is necessary to arrange the feed and clearance of material to and from the point of use in such a manner that the production process is not interrupted, even momentarily. A typical example of this need is found in the modern press shop. The maximum output of any given press if loaded to power capacity is governed by the press crank times. A loss of one stroke of the press can quite easily mean the loss of one component, and provision should be made to avoid this.

Where overhead conveyors for feeding and clearing the press is not practicable, other methods have been used. A floor conveyor is probably one of the simplest alternative methods. With it the operator can quickly exchange a full container for an empty, or vice-versa, without loss of time or undue expenditure of energy. In such an application it is usual to place the roller conveyor alongside the press or machine so that the trucker can take from one end and feed on to the other. For a similar function Vauxhall Motors Ltd. have developed a rotary table so that as one stack of metal is exhausted, the operator merely rotates the table to bring another stack into the most convenient operating position.

Machine tools and handling costs

Where continuous flow production is the aim, the planning engineer endeavours to break down the total manufacturing time to the maximum number of roughly equal time elements in order to obtain the planned volumes at the most economical cost rate. The ideal condition is that where each piece of processing equipment is fully loaded over the manufacturing period. It must, however, be understood that the greater the number of elements into which a manufacturing process is broken down, the greater will be the frequency with which the part is handled, either manually or mechanically. In fact, a stage could be reached when the movement and handling time could exceed the actual manufacturing time.

Now in order to obtain the maximum degree of flexibility and adaptability, British industry in the past has relied to a major degree on the general purpose machine tool. Frequently, this has resulted in the production of a number of different components on one machine, and often this has caused related production activities to become divorced from one another. As a consequence the manufacturing times have become unbalanced, the flow of production has been interrupted and the material pipe-line has been extended.

Nowadays, with the ever-increasing demand for higher output, especially in the motor car industry, the pendulum is definitely swinging in the opposite direction. Managements are beginning to realize that the general purpose machine tool, although having the greatest flexibility factor, does not always produce the most economical product. It is the recognition of this and of the enormous savings possible through a good material handling set-up that is forcing the trend more and more towards the installation of special purpose machine tools.

The development is interesting, from the days when one machine drilled one hole, through the stages of multi-head and index fixtures, to the present-day transfer machines. Each of these developments tended to reduce the amount of material handling. A transfer machine is in effect a battery of machines performing a multiplicity of operations, and electronically coupled to give straight-through flow. On such a machine the handling element of loading and unloading is completely eliminated except at the first and final operations. As a result related activities are co-ordinated, and inter-operational and inter-departmental movement is considerably reduced.

This gives not only a marked diminution of horizontal movement of "in process" material, but also a reduction in the amount of stock necessary to fill the pipe-line. Fortunately, the desirable element of flexibility in lay-out is not unduly impaired. In fact, the latest transfer machines are designed to facilitate the rapid removal or inclusion of individual units in order to accommodate production modifications.

What effect transfer machines may have on handling costs is disclosed by an analysis made by Vauxhall Motors Ltd. On their present cylinder block line, which operates on a flow basis, the components pass from machine to machine on a gravity roller conveyor running parallel with the machines. The main handling element on this line, transfer of the components between the conveyor and the machines by means of hand hoists costs £9,000 in direct wages. On the new line, which includes transfer machines and other modern equipment, there will be an immediate saving of the greater part of this £9,000.

Savings in material handling costs are not confined to those gained through the use of improved production machines. In most factories much time and energy are expended in handling salvageable by-products. In a modern factory, the economical handling of swarf, scrap metal, waste oil, wood offcuts and such like, is important. This aspect alone justifies a separate study.

Once again what can be done may be illustrated by recent Vauxhall developments in handling scrap sheet metal. Previously, three baling presses were used to deal with press shop scrap. The total volume involved was in the order of 1,200 tons per month. Although the balers were in almost continuous operation, they had insufficient capacity to cope with the complete tonnage.

In all, twenty men, scrap handlers, truckers and baler operators were employed in this activity. In addition, there was a loss of revenue in the order of £4,500 through the inability to bale all the scrap. Now, one triple ram baling press equipped with a pre-loading hopper and automatic bale ejector has been installed in place of the original three presses. This press has sufficient capacity plus adequate reserve to deal with the total tonnage. Through its use there is a reduction in personnel of 12 men and a total annual saving of rather more than £10,000.

Industrial Electric Motors

FACTORY maintenance engineers in general, and electrical engineering staff in particular, will find a wealth of useful information in the latest edition of Industrial Electric Motors. This publication is produced in a size that is convenient for carrying in the pocket. Its intention is to supply in a succinct and convenient form all the necessary information for the installation and maintenance of electric motors and generators. This intention is admirably fulfilled. Copies, price 1s 6d each, may be obtained from the compilers, Higgs Motors, Ltd., Witton, Birmingham, 6. (1900)

Jobbing Equipment

Black and Decker Ltd., Harmondsworth, Middlesex, have recently introduced a comprehensive jobbing equipment based on the electric hand drill. The power unit of the outfit is a small, but efficient electric hand drill, and a low priced but accurate drill stand is marketed, which instantly converts the hand drill for use as a drilling machine. A further adaptation consists of a small bench stand into which the hand drill fits and is held, for use as a grinder, buffer, or polisher.

Termed the "Handy-Utility," a feature of the outfit is, of course, its low price, and it is designed for general use in farm workshops, small garages, builders shops, and so forth where, for occasional usage, any considerable outlay is hardly justified. The outfit, apart from the drill stand, is marketed in a stout metal box, which also houses a set of high-speed drills, steel wire brush wheel, grinding wheel, buffing wheel, etc. As a Black and Decker product, the quality throughout is excellent and while each of the items can be purchased separately, the complete range of tools is priced at under £14.

CONTOUR MACHINING

A Review of a High-duty Swiss Copying Lathe

MULTI - diameter components may be machined in several different ways. In the determination of the method to be employed for any specific piece, several factors must be considered. Of these, the most important are (1) the size of the component and (2) the quantities to be produced. For example, small parts required in large quantities can frequently be most economically produced on multi-spindle automatics. Such machines, however, can accommodate only relatively small components. Nor are they generally suitable for use on small batches, since the tooling is usually such as to require a long setting-up time.

When components not suitable for multi-spindle automatics are to be produced, the planning engineer is faced with a choice of machines on which the job can be done, and capstans, vertical autos, multi-tool lathes and copying lathes are all employed for multi-diameter work. Once again several factors influence the final decision. It is not possible to say that any one type of machine is superior to the others in all circumstances. It may, however, be said that for multi-diameter work, even if only relatively small batches are machined, the planning engineer should certainly give close consideration to the possibility of using a copying lathe. These notes deal with the KDM copying lathes developed by George Fischer Limited, Schaffhausen, Switzerland, which are now used in many factories connected with the automobile industry.

Before discussing the features that are peculiar to KDM lathes, it is advisable to consider briefly the advantages of copying lathes in general. The first outstanding advantage is the simplicity of the tooling. For nearly all applications only one single point tool is used, no matter how many different diameters have to be turned. There is, therefore, the utmost economy in tool costs. Secondly, the copying principle employed gives the greatest possible simplicity in tool setting. Not only is there only a single tool that has to be set, but the setting is made in relation to only one length dimension and one diametral dimension. Once the tool is set to these dimensions, all other dimensions are automatically produced in the correct relationship.

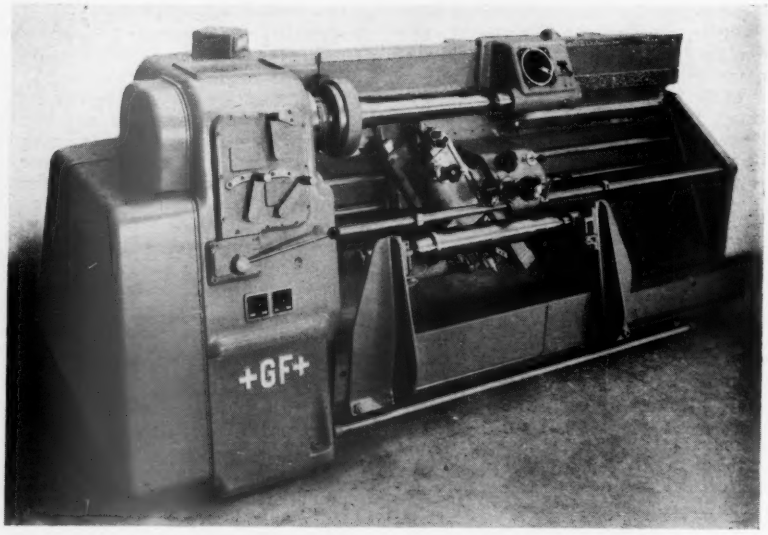


Fig. 1. Model KDM Copying Lathe (George Fischer Limited).

KDM copying lathes

Since these lathes have been specially designed to give high output rates, they have the power and rigidity necessary to allow full advantage to be taken of the metal removing properties of cemented carbide tools. They are available in four sizes with maximum turning lengths of 28, 40, 60 and 100in. The two smaller machines have a maximum turning diameter of 8⅞in and are powered by 18 h.p. motors while the two larger have a maximum turning capacity of 14in and are driven by 23 h.p. motors.

Actually, the smaller machines have a metal removal capacity of 6·2lb per minute and the larger machines of 8·4lb per minute. On a machine of conventional design, the disposal of chips accumulating at such rates can be a serious problem. They would probably necessitate the stopping of the machine while the chips were cleared away. In addition, the chips leave the stock at high temperature, and unless there was very good guarding, the heat and volume of chips could have a serious adverse effect on some of the working parts of a machine of conventional design.

On the KMD machines, a vertical bed is employed, and the chips have a completely uninterrupted fall to a chip tray beneath the machine. There is, therefore, no danger that the chips will damage any working surface. Furthermore, the chip tray can be withdrawn from the back of the machine without interfering with machine operation. In applications involving high rates of stock removal, this capacity for the removal of chips without stopping the machine may add from 8 to 10 per cent. to the effective

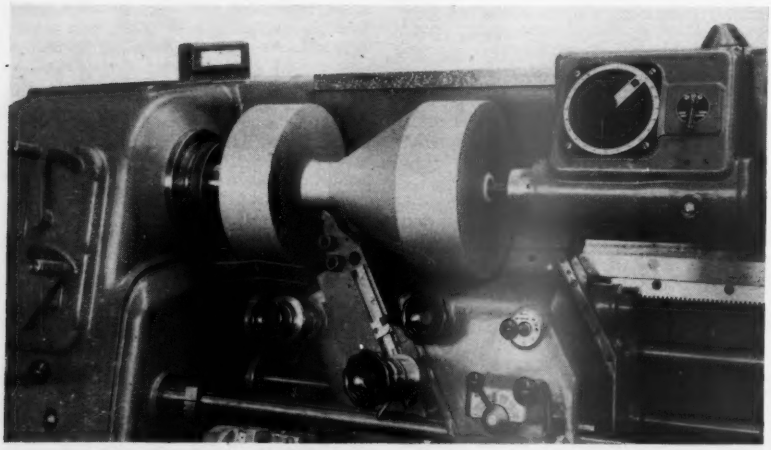


Fig. 2. Maximum copying capacity with one template and one single point tool.

working time with a proportionate increase in output.

Drive to the headstock spindle is taken through a close ratio gear box. On the smaller machines there are 12 spindle speeds ranging from 140 to 1800 r.p.m., and on the larger there are 18 speeds ranging from 75 to 1,400 r.p.m. A clutch and brake are mounted in the drive to reduce starting and stopping times to a minimum. An ammeter mounted on the headstock indicates the load on the main motor. For economical working, it is usually best to have the motor working at or near its full rated capacity. Incidentally, the motors are designed to run at 50 per cent. overload for 30 seconds. This overloading can be repeated indefinitely so long as the period between overload conditions is sufficient to allow the motor to cool down. The overload capacity can be an important factor in economical working. Obviously, in machining several different diameters and faces there must be variations in the load on the driving motor. Actually, the greatest load occurs in machining a face, generally over a period of three or four seconds. Therefore it is possible to carry out the greatest portion of the machining at a load approaching the full rated capacity and have an overload for three or four seconds for facing.

A motor-actuated tailstock with a built-in live centre is employed. It incorporates provision for automatically compensating for expansion of the workpiece due to heat. As these machines may be used for heavy parts, the tailstock is arranged for operation by a foot bar switch so that the operator may have both hands free for handling the work. The tailstock incorporates a selector lever with three precisions, neutral, advance, and retract. To unload the machine, the operator moves the selector lever to retract and then presses the foot bar switch. As soon as the pressure is removed from the foot bar the lever automatically returns to the neutral position. This arrangement completely eliminates any danger that the tailstock centre shall be retracted through inadvertent movement of the foot bar while the machining is running.

Provision is made for using either a plate template or a prototype as the master. A prototype is often convenient for simple work-pieces or for a non-recurrent batch. As the contact pressure of the tracer against the master is light, plate templates can be made from thin unhardened steel. If, however, a comparatively large quantity of one component is to be made, it is advisable to flame harden the portions of the template that represent faces on the component. The plate template or prototype holder is located in full view and within easy reach of the front of the machine. It incorporates a simple device with setting scales for horizontal and vertical adjustment to allow change-over to be carried out quickly and easily.

The copying slide is hydraulically operated and is mounted on the carriage at an angle of 60 deg. to the work spindle axis. When the tracer contacts a shoulder on the template, it is possible to reduce the feed rate to one-half. On standard machines this is effected manually by movement of a lever, but it can be arranged for automatic operation through trip dogs on the plate template.

Drive to the carriage is taken from the main spindle by shaft through the feed gearbox and a reverse gear. The smaller machines have nine feed rates ranging from 0·0022in to 0·0177in per spindle revolution, while the larger machines have eight feed rates from 0·0018 to 0·020in per spindle revolution. On all machines rapid traverse is at the rate of 100in per minute. The feed and rapid traverse movements can operate in both directions. At the end of the operation controlled by the template the tool is automatically retracted and the rapid traverse motion is engaged to return the carriage to the starting position.

The simplest form of copying is that of producing an external form with the diameters increasing progressively in the direction of feed traverse. When this is the case, the finished form of the component can be produced in a single pass, providing that the amount of metal to be removed is within the capacity of the machine. Provision is also made for carrying out a roughing and finishing cut at one setting and using only one template. On the standard machine the change from roughing to finishing cut is made manually through a depth of cut device, but where a machine is to be used for large quantity production, a special device can be fitted to give a completely automatic cycle.

Because of the angle at which the tool operates, it is not possible to produce a right angle shoulder where a larger diameter comes before a smaller diameter in the direction of travel. The actual form that can be produced is shown in Fig. 2. If a square shoulder is required, it can be produced by using a duplex tool holder and two templates. With this arrangement one tool cuts on the forward stroke of the machine and at the end of the traverse the feed is reversed and the second tool cuts on the return stroke.

Internal copying can also be easily carried out. For such operations, a boring bar is mounted in the copying slide. There are, of course, the same limitations in internal copying as in external copying, and once again it is possible to use two templates and two tools for copy turning opposite shoulders.

The most recent additional equipment for use with these machines is a power-operated facing head. This is for use in rough machining deep flanges. In facing a flange the load on the tool is much greater than in straight turning, for equal depth of cut. If however, the power-operated facing head is used to take a roughing cut, the copying tool takes only a light finishing cut on the face and there is no danger of overloading. Sidney G. Jones, Ltd., 8 Balham Hill, London, S.W.12, are the agents for these machines.

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

Resistance Welding

By A. H. Hipperson, B.Sc. (Eng.), A.M.Inst.W., and T. Watson, M.Inst.W.

London: Published for *Welding* by ILIFFE AND SONS, LTD., Dorset House, Stamford St., S.E.1. 1950. 278 pp. $5\frac{1}{2} \times 8\frac{1}{2}$. Price 21s.

The importance of welding in modern production grows, and an increasing number of books on the subject are now becoming available. Each deals with some particular aspect of the subject, and this one is different from most in that it treats with the processes of resistance welding from the mass production angle. Welding procedure lends itself admirably to mass production technique, and there is considerable scope amongst welding departments and manufacturers generally, for improvements in these directions. The growing appreciation and use of resistance welding in industry is largely due to its suitability to modern mass production. The authors have had considerable practical and theoretical experience of welding. Mr. Hipperson is chief development and liaison engineer to the British Welding and Research Association and Mr. Watson is works manager at Walton and Brown, Ltd., and he was formerly welding engineer to Joseph Lucas, Ltd. The book therefore embodies both the experience of research and practical mass production welding.

Considerable space is devoted initially to selection of the correct process. It may be possible to use spot, seam, projection or butt welding, or even hot riveting may be convenient, but the authors emphasize that it is more important to decide what equipment is necessary after deciding the process that lends itself best to mass production, than to use the available equipment which might reduce working efficiency if it entails the incorrect process of welding.

The various processes are dealt with individually. In each case the process is divided into principles, welding machines

or equipment, electrodes, and jigs and tools required. Many illustrations of typical jigs are included in each case. The use of multi-station indexing machines in conjunction with mass production projection welding is discussed.

A chapter is devoted to the control equipment for use with resistance welding machines. This apparatus is of a special nature owing to the fact that high currents have to be switched on and off very frequently and with such precision that the duration of current flow shall be substantially constant. The principles of operation of trip switches, contactors, reducing valves and details of heat control by phase shift are all dealt with in a manner that will be readily followed by welders having only a fundamental knowledge of electricity.

A number of tables included in the chapter on recommended machine settings deal with mild steel, stainless steel, steel sheet, steel tubes, etc., for flash, seam and projection welding. The section on testing and inspection of welded joints first deals with design of joints and here are reviewed the various types of resistance welds, with notes on the strength that is to be expected from them. Under spot welding tests are found static tension-shear tests, tension-shear specimens under load, static tension tests, impact tests, fatigue tests, and shop tests. Some brief notes are included on tests for projection, seam and flash welds.

The final chapter is devoted to resistance welding costs, and tables in this section include some in connection with labour and weld energy costs involved in spot and seam welding. The importance of ensuring high production speeds in welding are emphasized by the fact that even at a good speed of 40 welds per minute, the cost of the operator's time may exceed three times the sum total of all other direct costs. The book is well written and well illustrated.

Engineering Precision Measurements

By A. W. Judge, A.R.C.S., D.I.C., Wh.Sc., A.M.I.Mech.E., Assoc. F.R.Ae.S.

London: CHAPMAN AND HALL, LTD., 37, Essex St., W.C.2. Second Edition. 1950. $6\frac{1}{4} \times 8\frac{1}{2}$. 363 pp. Price 30s.

As its title conveys, this book covers engineering precision instruments, as distinct from physical, scientific or experimental measuring equipment. It describes the simpler methods of linear and angular measurements covering the use of the vernier and micrometer, and then describes the plain and limit gauge methods. The checking of this equipment is then dealt with before discussing the use of absolute standards of measurement employed for checking the other measuring apparatus.

The use of measuring equipment is a vital factor in modern mass production, and obviously, precision becomes more and more important. Equipment must also be made as foolproof as possible so that accurate work can be carried out by semi- or unskilled labour, and the time taken in checking parts must be reduced to a minimum. Where tolerances allow, slow methods need not be adopted, such as when two or more differing measuring instruments must be handled by the same workman in checking one small part. Provision must be made for ensuring accuracy over long periods of use and motion study ensures that the design of instruments shall be such as to be quickly and untiringly handled by operatives.

Thus, the subject is a considerable one, and in this book the author deals with every measuring instrument used in mechanical engineering from a simple vernier to the latest types of surface finish recording equipment. Included in this wide range are limit, profile and slip gauges, thread gauging and measuring equipment, measurement of angles, comparators and other optical methods of measurement, and a number of special methods and appliances including the profilograph, continuous metal strip gauging machine, camshaft testing apparatus, and gear tooth callipers.

Finally, this second edition contains a chapter of fifty pages dealing with the latest developments in precision measurement appliances that have come into more general use since the first edition of this book in 1944. Here will be found descriptions of the Societe Genevoise type MUL measuring machines, used for checking working gauges used in mass production inspection. New type slip gauges, thread gauges and apparatus for making internal measurements are incuded, although in many cases, only brief descriptions can be given.

Recent information on gear testing appliances will be found and the N.P.L. device and the David Brown tangent micrometer are mentioned. The use of the Hilger Angle Dekkor for micrometer testing is described, one of the many applications of this valuable equipment, and the latest Inferometers and surface finish recorders are briefly dealt with. The book is well illustrated and the text should prove clear to any user or student of precision equipment.

Metal Finishing Handbook

By P. A. Cartwright, A.M.G.I., A.M.I.E.I., L.I.M.

London: BLACKIE AND SON, LTD., 66, Chandos Place, W.C.2. 1950. 216 pp. $5\frac{3}{4} \times 8\frac{1}{2}$. Price 15s.

Corrosion and rust prevention of metal surfaces are increasingly important subjects, especially so in view of the fact that a very great number and variety of products are being sent from this country to all parts of the world. They must, therefore, withstand storage before and after despatch, different climatic conditions and almost always meet conditions that promote surface attack. Frequent loading and unloading, docks storage, subjection to differing atmospheric conditions, varying temperatures from arctic to tropical, and so forth, cause rapid deterioration of parts that have no protective coating. These troubles were met in full severity during war time, and the lessons learned then, have been most valuable in connection with our export drive. Materials must arrive at the premises of a foreign customer in exactly the same condition as they left the manufacturer's factory. This applies particularly to all motor vehicles and includes their every component part.

Preventive coatings may be either permanent or temporary and this book sets out, as simply as possible, the working processes used in providing permanent metal protective finishes. It first deals with the pretreatment of metal surfaces by grinding, polishing, burnishing, chemical cleaning, pickling, scratchbrushing, and stripping. It then describes electroplating, and the electrodeposition of metals such as nickel, copper, chromium, cadmium, zinc, lead, etc. It also covers decorative plating processes such as gold, silver, rhodium, palladium, cobalt, tungsten, bronze, silver cadmium, etc.

A chapter is devoted to metal colouring by bronzing, oxidizing, blueing, lacquering, colouring of silver, gold, tin, nickel, aluminium, etc. Brief descriptions are included of phosphate and allied coatings such as parkerizing, bonderizing, granodizing, chromodizing, etc. The references to dipped and allied metallic coatings include galvanizing, tinning, cementation processes, sherardizing, sand-blasting, etc., with details concerning the final treatment, adhesion, porosity and applications of these processes.

The final chapter covers the testing of these coatings under adhesion, thickness, porosity, and corrosion methods for quantitative evaluation of various protective coatings are given and these deal with nickel, silver, gold, copper, lead, cadmium, chromium, tin and zinc. The appendices deal with Plating Shop Control and Specifications, Laboratory Technique in Chemical analysis, useful constants, First Aid, Glossary of Terms, and a valuable Bibliography.

The many subjects treated in this book would, of course, provide matter for several volumes, each larger than this one. The author has purposely omitted most of the theory behind the many processes, and the reader wishing to follow this may refer to the bibliography. Detailed description of the plant mentioned is not included, since, as the author states, this is in fact always provided by the manufacturers of such specialized equipment. Details of mass production methods are also excluded.

Workers in the metal finishing industries will be interested to have concise, up-to-date, information concerning those processes allied with their own. Manufacturers who buy materials and products that have been treated for corrosion prevention or merely for decorative purposes, will be interested in having practical explanation and description of the processes involved. Chemists will be interested in most of the book but especially in the detailed compositions of the many plating solutions since these should prove valuable if they ever have to analyse these preparations.

Mechanical World Year Book

Manchester: EMMOTT & CO., LTD., 31, King Street West, Manchester, 3. 1950. 4×6. 360 pp. text, 268 pp. manufacturers' announcements, directories, etc. Price 3s 6d.

In its sixty-third year of publication, this little volume continues to provide a wealth of information at low cost. The principal addition this year is a section devoted to the continuous flow gas turbine, by W. R. Thomson, B.Sc., A.M.I.C.E. This sets out the basic thermodynamics, adiabatic changes, the constant pressure cycle, specific heat, bladed turbo-machines, and gives details of some plants constructed. This section will need extending yearly to keep it up to date. Much of the tabular matter has been rearranged and related subjects have now been brought together so that quicker reference to detailed information is possible. A contents page has been included in addition to the index.

The text includes notes on most subjects of interest to designers and general mechanical engineers. Many of the sections have been written by experts in their respective fields. Some of these

cover belt conveyors, fuels and lubricating oils, machine tools, compressors, the light alloys, plastics, steam boilers, high-speed tools, toothed gearing, steam turbines, etc. Many will find the numerous tables as valuable, if not more so, than the text since they cover a wide variety of subjects including price equivalents, metric conversions, weights and measures, steam and thermometric tables, pipes and tubes, screw threads, tolerances and fits, strength and properties of plates, bars, etc., weight calculating tables, etc., etc. The numerous manufacturers' announcements, which are also classified in a buyer's directory, increase the value of the book to a wide circle of readers.

The Electrical Equipment of Automobiles

By Stanley Parker Smith, C.B.E.

London: CHAPMAN & HALL, LTD., 37, Essex Street, W.C.2. 1949. $4\frac{3}{4} \times 7\frac{1}{4}$. 331 pp. Price 12s 6d.

Since the first edition of this work appeared in 1927, considerable additions and some revision have been carried out from time to time. The present edition, the fifth, includes some new material whilst retaining all that might be concerned with cars in present use, including the old ones. For example, there is a section dealing with the magneto which though practically obsolescent on motor cars is still to be found on some motor cycles, some commercial vehicles and tractors, and on aero-engines.

The book is primarily of value to the electrical department of service stations. It covers all the equipment to be found on a motor vehicle, including horns, direction indicators, windscreen wipers, car heaters, speed indicators, gauges, press-button window winders, radio-interference suppressors, etc., although only very brief descriptions of most of these are given. The main chapters in the book deal with the battery, starting motor, lighting equipment, automatic cut-out and charging generator, the ignition system, and faults locating. A chapter giving fundamental principals of the internal-combustion engine has been included, since the electrician must have this knowledge if he is to understand the working of the ignition system.

Two appendices have been included in this edition. The first covers laboratory instruction for automobile classes. Here, experiments which might be employed as instruction to students deal with batteries, starter motors, dynamos, ignition equipment, regulators, lighting circuits, cut-outs, and accessories. The second appendix deals with calculations and consists of six problems with fully worked solutions dealing with demands made on batteries during starting and manœuvring. The book is easy to read and understand and it is illustrated with line drawings and graphs.

Diesel Maintenance

By T. H. Parkinson, M.I.Mech.E. Edited by Donald H. Smith, M.I.Mech.E., Assoc. Inst.T.

London: Published for *Motor Transport* by ILIFFE AND SONS LTD., Dorset House, Stamford Street, S.E.1. Third Edition. 1950. $4\frac{1}{4} \times 6\frac{3}{4}$. 243 pp. Price 7s 6d.

This book is concerned with the practical maintenance and servicing of diesel engined vehicles and is a revised and up-to-date edition. Nearly half of it is devoted to injection equipment, and certainly this is the most important part of diesel equipment from the practical operator's point of view. This section of the book is divided into three parts: (1) routine maintenance, sequence of adjustments, filters, sprayers, fuel lift layout, (2) pump adjustments, timing, phasing, governors, controls, (3) calibration, overhaul, specialization, experimental work and modifications. An injection timing chart covering most modern road diesel engines, and a table of injection pump flow settings are included.

The rest of the book deals with engine maintenance and again some valuable charts and tables are included covering such items as valve spring details (length, pressure, etc.), piston ring clearances, big-end and main bearing clearances, etc., for several popular diesel engines. Possibly the book would be made still more valuable, if these details were extended to cover all diesel engines fitted to vehicles during recent years.

Chapters are devoted to adjustment, bench and road testing, garage and workshop layout. The final chapter, headed "Useful Items" is an appendix of practical details concerning various facts pertaining to individual makes. For example a tip is given to facilitate removal of the cylinder head from the Leyland engine, there are some notes on "masked" inlet valves, timing diagrams for some popular engines, valve seat angles, etc. Information of this kind is always extremely valuable and it is always a pity that there is not more of it. A little practical knowledge is sometimes worth a mass of technical data, and practical hints and tips are appreciated by practical workshop personnel for whom this little book is intended. It is assumed by the authors that the reader has a working knowledge of the principles of diesel engines.

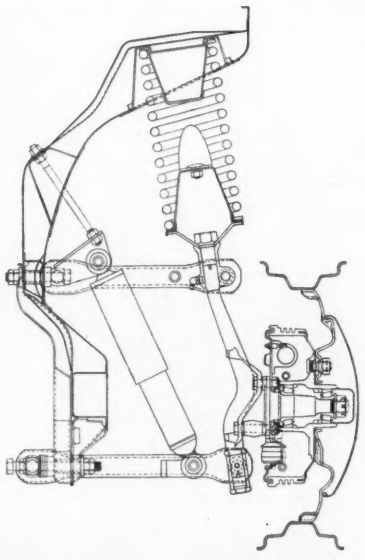
THE NASH RAMBLER

Unusual Features in a New Medium-sized American Car

SEVERAL features unusual in American car design are embodied in the new medium-sized car introduced by Nash Motors. According to *Automotive Industries*, the Nash Rambler has a wheelbase of 100in, a shipping weight of 2,400 lb and is powered by a six-cylinder side valve engine of $3\frac{1}{8}$in bore by $3\frac{3}{4}$in stroke, developing 82 b.h.p. at 3,800 r.p.m. and having a compression ratio of 7.25 to 1. The most interesting technical feature of the car is the independent front suspension layout, shown in the accompanying illustration.

In this design wishbones and coil springs are employed, but whereas in the customary layout the coil spring seating is formed with the lower wishbone and with an abutment provided by a rigid front frame cross member, in the Nash Rambler the coil spring is mounted directly above the stub carrier and its abutment is formed on the body structure. It has thus been possible to eliminate the usual heavy front cross member, and the front end of the frame can therefore be designed to accommodate the engine well forward.

It is pointed out that with this suspension arrangement the front springs take practically direct wheel loads and the suspension linkage, in particular the lower wishbones, is no longer subjected to vertical and bending loads since its duty is restricted to the maintenance of wheel alignment and thrust loads. It is claimed that the linkage can therefore be made lighter and that since vertical loads are no longer transmitted by the linkage, the inner pivots of the wishbones need not be insulated from the body. This type of construction is basically similar to the Voran front suspension and the Daimler "Scout" armoured vehicle.



Nash Rambler front suspension.

Another departure from established Nash practice is the adoption of semi-elliptic rear springs and Hotchkiss drive, recent Nash cars having employed coil springs in conjunction with torque tube drive. Direct-acting dampers are fitted, mounted in "reverse sea-legs" manner, that is to say, the upper mounting points are further apart than the lower.

So far, only a convertible body has been announced and it follows a type of body that has proved particularly popular on Continental cars, that is to say, the fabric top which is retracted by an electrically driven drum mounted behind the rear seat, is supported by four intermediate cross members guided by the side rails of the body structure, which, of course, provide the necessary stiffness that would otherwise be lacking in a conventional drophead design. (1896)

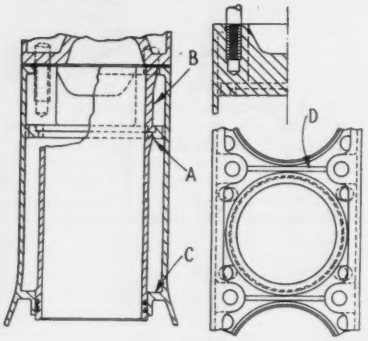
CURRENT PATENTS

A Comprehensive Review of Recent Automobile Specifications

Cylinder construction

TO minimize the risk of distortion of the bore, as a consequence of tightening the cylinder-head bolts or of uneven cooling, this wet-type liner is supported in the block on a shoulder spaced from the upper face and the upper end is wholly out of metallic contact with the block. The side walls of the block are bridged by platform A formed with annular seatings accommodating the support shoulders of liners B and by platform C having flanged apertures to receive the lower ends of the liners.

At their upper ends the side walls have inturned flanges and, intermediate of the liner apertures, cylinder head stud bosses connected by transverse web D. The upper faces of liner and block lie on the same plane and either a common or individual gaskets are used to seal the junction with the cylinder head. Preferably, a relatively soft gasket is used for the block and relatively hard rings for the liners. *Patent No.* 615045. *Twin Coach Co. (U.S.A.).*

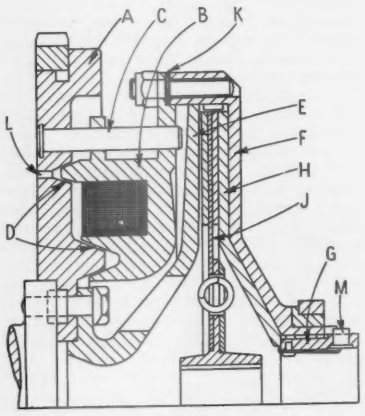


No. 615045.

Electromagnetic clutch

FOR a given excitation the axial force developed by an electromagnetic assembly varies to a prohibitive extent if the air gap is permitted to depart from a relatively narrow range. In practice this calls for fairly frequent adjustment and the object of the invention is to facilitate this operation.

The flywheel A forms the armature for the channel-section, annular electromagnet B axially slidable on drive pins C. Outer and inner air gaps D are constituted by tapered annular recesses and complementary projections to secure maximum variation before adjustment becomes necessary, as detailed in an earlier patent, No. 607820. The inner clutch face is an annular steel plate E rigidly secured to the flywheel. A dished cover plate F secured to the electromagnet has a screw—threaded axial bore accommodating a screwed bushing G to provide an adjustable abutment for the presser plate H, the periphery of which engages driving splines in the rim flange of the cover plate. A driven shaft (not shown) extends through this bush and carries a conventional clutch plate J.

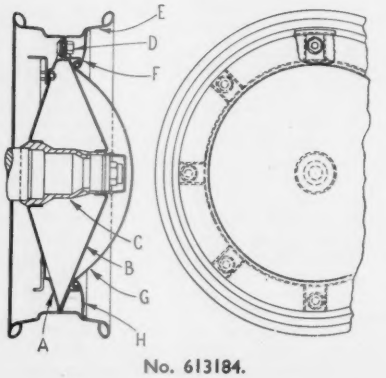


No. 617406.

The magnet is designed for a working throw of 0.0625in and an allowance of 0.0625in between adjustments, so the bush screws into the cover plate with 16 threads per inch. Initial setting of the gap is by means of shims K. Wear is ascertained by inserting a depth gauge marked with the permissible limits of the air gap through a hole L in the flywheel. When, as a result of wear of the friction surfaces, the air gap has been reduced to the minimum a spanner is applied to the protruding hexagon end of the bush G and depresses the spring-pressed locking plunger M. Thereafter, the engine is cranked exactly one revolution and the spanner removed permitting the assembly to be locked in its adjusted position. *Patent No.* 617406. *A. C. Sampietro and D. R. Robertson.*

Divided-rim wheel

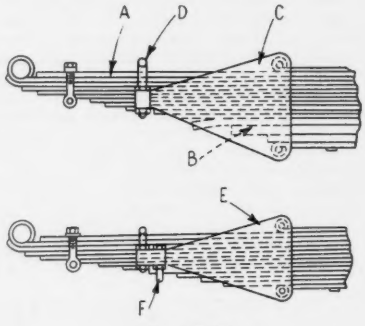
IN this wheel the body comprises discs A and B with flanged central apertures which are secured, preferably by brazing, to the hub C. The discs converge radially outwards and contact at the periphery. Disc A is extended to form the inner half of the drop-centre rim. Angularly spaced opposed pockets pressed in the two discs meet in the approximately central plane of the wheel and receive the bolts D, which are permanently secured by staking. The demountable half E of the rim has complementary pockets and is also formed with a resilient inner flange F over which can be sprung the nested beads of covers G and H. The brake drum is bolted to an annular boss formed on disc A. *Patent No.* 613184. *Kelsey-Hayes Wheel Co. (U.S.A.)*



No. 613184.

Suspension of heavy vehicles

SERVICE requirements of some heavy vehicles, particularly but not exclusively double-decker omnibuses, involve capacity for safe tilting under load. Usually some compromise in spring design or construction is necessary to enable the springs to be strong or rigid enough to secure the desired angle of tilt yet flexible enough to give riding comfort for passengers. In this construction the object is achieved by a pair of rebound check stops rigid with respect to the axle and thus avoiding the use of slings mounted on the vehicle frame.



No. 617208.

In one method a spring A operates in conjunction with a heavy auxiliary spring B which serves as a bump damper cumulative in action due to the graded clearance between the upper plate and the adjacent spring leaf. Rebound damper action is obtained by two pairs of triangular side plates C rigidly anchored at their bases to the top and bottom spring plates. The apices of each pair are joined by a U-bolt D which straddles the spring. An appropriate clearance can be secured by adjustment of the U-bolt nuts.

In an alternative construction, the auxiliary spring is omitted, stronger side plates E are provided and additional U-bolts F are fitted to serve as bump dampers. A third form (not shown) has a stiff beam of a length less than the total length of the spring clamped to the axle and from the ends of this beam a U-bolt goes round the spring. Apart from controlling the tilting angle of the vehicle these constructions will also limit the degree of roll when cornering. All possess the advantage of being applicable to existing vehicles with a minimum of constructional alteration. *Patent No.* 617208. *The Associated Equipment Co., Ltd., and A. A. Houston.*